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# Performance comparison of enhanced steam condensers.

Bourgeois, Neil Charles.

Massachusetts Institute of Technology

# PERFORMANCE COMPARISON OF ENHANCED STEAM CONDENSERS

by

NEIL CHARLES BOURGEOIS
LIEUTENANT, U.S. NAVY

B.E., State University of New York,
Maritime College
(1975)

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ABSTRACT

The objective of this project is to investigate Vertical Steam Condenser Design in an effort to develop low weight, small volume condensers utilizing doubly enhanced (coolant and steam side augmentation) tubes. An existing vertical condenser analytic model for external axial fluted tubes was improved to include internal (coolant side) heat transfer augmentation. A computer program (VERTCON-1) was developed to be used as a preliminary design tool for sizing the total condenser. A performance comparison was conducted between enhanced vertical condensers, horizontal smooth and horizontal enhanced tube condensers.

This analysis has clearly demonstrated that significant reduction in condenser weight and volume can be obtained for certain applications using enhanced vertical tube condensers. Also, enhanced vertical tube condenser performance compares well with present designs for enhanced horizontal tube condensers.

Thesis Supervisor: Warren M. Rohsenow

Title: Professor of Mechanical Engineering

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# I. INTRODUCTION

The current practice in Marine Condenser Design is to use smooth horizontal tubes. In comparison the performance of horizontal tube bundles is generally lower than for a single tube, this deleterious bundle effect depends on the pattern of accumulated condensate drainage, and the number of rows of tubès in a vertical array. This could cause a significant decrease in condensing coefficient for large tube bundles. This sort of phenomena will not occur in vertical tube condensers.

The heat transfer on these vertical tubes can be further enhanced by grooving the external surface of the tube (Fig.1). Surface tension acting through gradients in the curvature of the liquid meniscus will draw liquid into the grooves. The condensate is drawn off the tips of the flutes so that only a thin liquid film remains on them. The reduced thickness of the condensate film at the flute tip dramatically increases the heat transfer coefficient.

Internal tube heat transfer augmentation is provided through turbulation techniques by the use of integral multiple-helix ridging (6) as shown in Fig. 2. Therefore, these tubes with both external axial flutes and internal helical ridging can be said to be doubly augmented. The internal ridging provides a cost-effective form of internal roughness, which results in a substantial decrease in waterside thermal resistance.

Condensation on fluted surfaces and the resulting enhancement of the heat transfer coefficient was first recognized by Gregorig (1954). It has been just till recently that this concept has been put into practical application. The experimental work that has been completed has clearly shown that the rate of condensation on fluted surfaces is several times

greater than that on smooth surfaces.

Presently work is being done in the determination of optimum surface geometry for vertical fluted condenser tubes. The basic principle for optimum performance is to make the thickness of the condensate film on the crests of the fluted surface as thin as possible, and to effectively remove collected condensate. From analysis of MORI et al (3) the basic controlling factors for optimum flute performance are:

1. sharp leading edge
2. gradually changing curvature of flute surface from its tip to the root
3. wide groove between fins to collect condensate
4. horizontal disc set to the tube to remove condensate (see Fig. 3)

Also, vertical tubes provided with longitudinally parallel tiny flutes is preferable because condensate film is made thinner over the widest possible region. As vapor condenses on the tube surface the grooves in the lower sections of the tube can fill up with condensate resulting in deterioration of condenser performance. The purpose of the horizontal discs is to remove condensate thus preventing the deleterious effects of this flooding condition. The spacing of the discs must not be too close together, since for narrow spacing of discs surface tension can also pull condensate on the discs up along the grooves causing the flooding condition. Condensate removal for the proposed vertical condenser design (Fig. 4) is accomplished by the tube support plates, where condensate is collected and removed to the bottom of the hotwell by downcomer drainage tubes.

The flute pitch can also affect condenser performance. Flutes with small grooves (small pitch) are easily filled with condensate resulting in reduced performance. A tube with flutes of too large of a pitch has a smaller number of flute tips, and performance suffers since the heat transfer

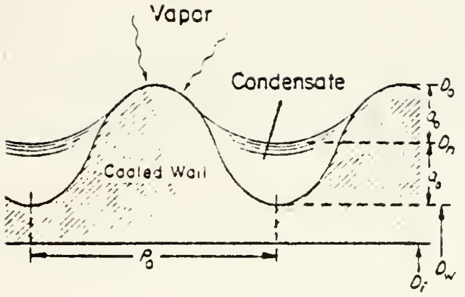



Figure 1.

Cross-Section of Vertical Fluted Condenser Surface

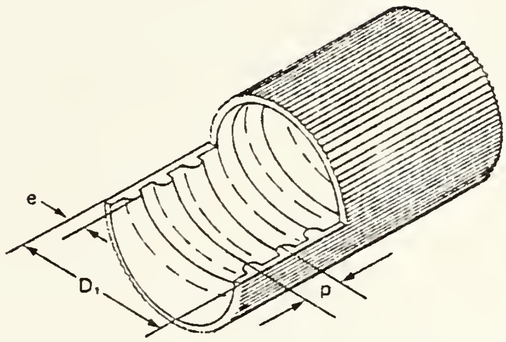



Figure 2.

Internal Heat Transfer Augmentation for Vertical Tube

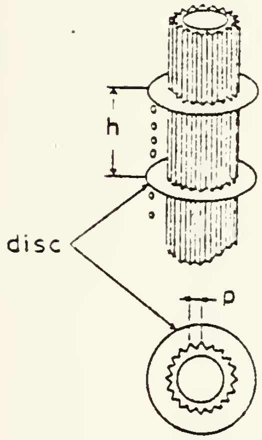


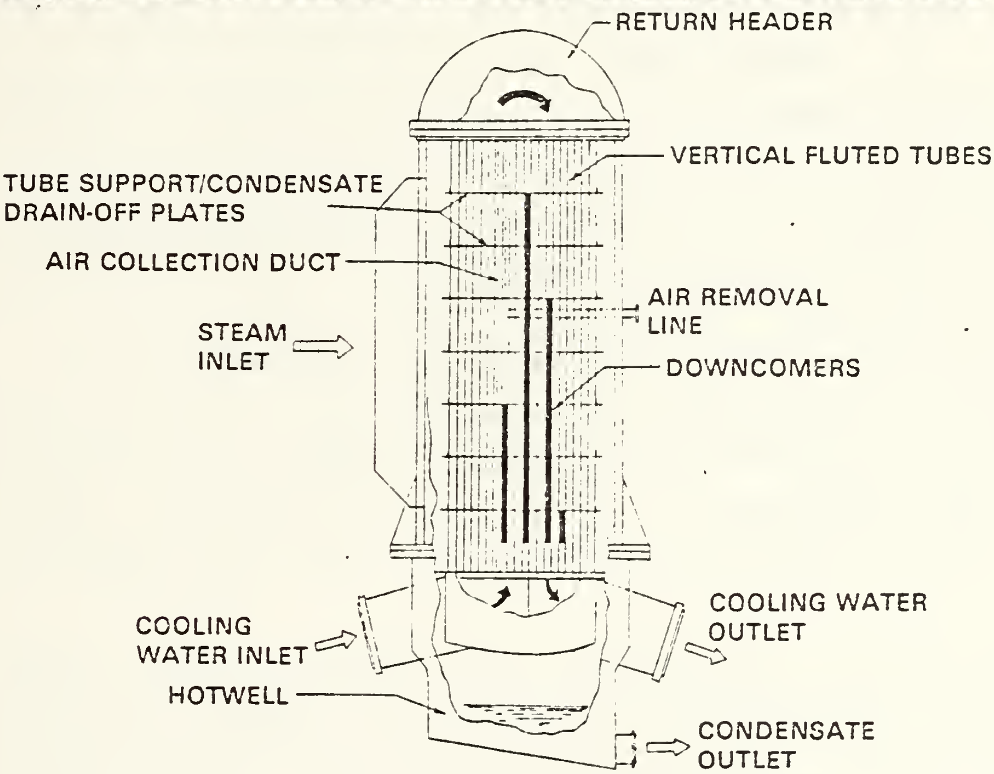

Figure 3.

Vertical Fluted Tube Fitted With Spaced Circular Disks for Condensate Removal

enhancement effect is not used to a great enough extent. Therefore, the condenser tube provided with flutes of optimum pitch for a given fluid has the best performance. The study conducted by MORI et al (3) has indicated that the optimum spacing of discs is smaller than 100 mm (3.94 in.) for water and for copper tubes with flute pitch on the order of .5 mm (.0197 in.) and flute height of .87 mm (.0343 in.).

FIGURE 4

# VERTICAL ENHANCED TUBE CONDENSER



The proposed vertical condenser arrangement for this investigation is shown in Figure 4. Steam enters the condenser shell along a major portion of the tube length. Steam lanes distribute the steam around the tube bundle with radial steam inflow into the tube bundle towards the air cooler section and air removal duct. Steam condenses on the vertical tubes, and the condensate collects in the valley of the tube fluted surface flowing down the tube length. The tube support plates have a dual purpose by also serving as condensate drain-off plates. The condensate is stripped off the tubes and collected on the tube support plates, and then it is removed to the hotwell section via downcomer drainage tubes. Improved condensate control is afforded by this condensate drainage and removal scheme. The tube support/condensate drain-off plates divide the condenser up into sections, where the full depth of tubes in each section see relatively fresh steam. The tubes will see fresh steam through the depth of the tube bundle since the tubes are not subjected to condensate indundation effects because of effective condensate removal and vertical orientation of the tubes. A continuous duct that runs along the total tube length is utilized for air removal.

Figure 4 shows a double flow cooling water circuit with the inlet/outlet header located in the bottom of the condenser. The condenser hotwell is located below and also encases the inlet/outlet header. This hot-well location adds to the total height of the condenser.

The possible condenser locations are either along side the propulsion turbine with side exhaust into the condenser, or located in line with the propulsion turbines with end exhaust into the condenser.

This investigation of vertical steam condensers will clearly establish that for certain condenser operating conditions significant reductions in weight and volume can be achieved. Performance comparisons were conducted with horizontal smooth and enhanced tube condensers. In some respects the vertical condenser will involve complications in design, fabrication and propulsion system integration, but the techniques required are well within the scope of present-day technology.

The objective of this study is to conduct a performance comparison between enhanced vertical steam condensers, horizontal smooth and horizontal enhanced tube condensers. The vertical condensers in this performance comparison were sized using the design routine established in this study, and compared with existing baseline horizontal condenser designs.

## II. HEAT TRANSFER CALCULATIONS

### Nomenclature

| | |
|---|---|
| $A$ | area ($ft^2$) |
| $a$ | amplitude of the flute (ft) |
| $Cp$ | specific heat (BTU/lbm°F) |
| $D,d$ | Diameter (ft) |
| $e$ | Helical ridge height (ft) |
| $f$ | Friction Factor[ $=(\Delta P/\rho)(D/L)(2g_c/V^2)$] |
| $G$ | Mass Flux (lbm/$ft^2$ hr) |
| $g_c$ | Gravitational constant (lbm ft/lbf $hr^2$) |
| $h$ | Heat transfer coefficient (BTU/hr $ft^2$ °F) |
| $h_{fg}$ | Latent heat of vaporization (BTU/lbm) |
| $K$ | Thermal conductivity (BTU/hr $ft^2$ °F) |
| $L$ | Length (ft) |
| $l$ | Lead of ridge (axial distance per 360° turn)(ft) |
| $m$ | Operand for friction factor equation |
| $Nu$ | Nusselt number |
| $P$ | Pressure (lbf/$ft^2$) or (in-hg-abs) |
| $p$ | Pitch of ridging or flute (ft) |
| $Pr$ | Prandtl number |
| $Q$ | Heat flow (BTU/hr) |
| $r$ | Operand for friction factor equation |
| $Re$ | Reynolds number |
| $Re_f$ | Flooding Reynolds number for flute[$= 4W_f/\mu X_L$]** |
| $R_s$ | Thermal resistance of scale (hr$ft^2$ F/BTU) |
| $R_w$ | Thermal resistance of tube wall (hr$ft^2$°F/BTU) |
| $St$ | Stanton number |
| $T$ | Temperature (°F) |
| $\Delta T$ | $T_{sat} - T_W$ (°F) |
| $\Delta T_{lm}$ | Mean overall temperature difference (°F) |
| $U$ | Overall heat transfer coefficient (BTU/hr$ft^2$°F) |

$U^+$ Dimensionless Velocity[$= U/U^*$] *

$U^*$ Friction Velocity$\left\{=\sqrt{\tau_0 g_c/\rho}\right\}$ *

V Velocity (ft/hr) or (ft/sec)

$W_f$ Flooding axial mass flow of condensate per flute (lbm/hr)

$X_L$ Half-perimeter length of flute (ft)

Subscripts

b Fluid at the bulk temperature (°F)

c Condensate

f Fluid or flooding

i Inside, or inlet

n Nominal

o Outside, or outlet

s Increment, or section

sat Saturation

w Wall

Superscripts

+ Dimensionless Parameter

Greek Symbols

$\alpha$ Height of the condensate in the center of the flute (ft)

$\gamma$ Fragment of $U_e^+$[$= -2.5\ln(2e/d_i) + 3.75$] *

$\lambda$ Dimensionless Group $= \dfrac{4\rho^2}{\mu^2}\, g_c\, \dfrac{(\alpha_0)^4}{X_L}$ **

$\mu$ Dynamic viscosity (lbm/hr ft)

$\rho$ Density (lbm/ft$^3$)

$\sigma$ Surface Tension (lbf/ft)

$\Omega$ Non-dimensional group +

$\tau$ Apparent wall shear stress

* Reference (6)
** Reference (4)
\+ Reference (1)

## II A. ANALYSIS SUMMARY

The rate of heat flow Q over an entire heat exchanger is related to the mean overall temperature difference $\Delta T_{lm}$ and the total heat transfer area A by the overall heat transfer coefficient U.

$$Q = UA\,\Delta T_{lm} \qquad (1)$$

From the analysis of externally ridged tubes it proves convenient to base the heat transfer coefficient U on the surface area of a smooth tube having an outside diameter $D_n$ equal to the diameter measured over the mid-height of the external flutes (see Figure 1 for external flute geometry).

$$U_n = \frac{Q}{\pi D_n L}\,\Delta T_{lm} \qquad (2a)$$

where $$D_n = D_W + 2a \qquad (2b)$$

The overall U depends on the resistances in series; between cooling water and the tube wall, within the tube wall, and between the tube wall and the working fluid. Fouling resistance on either side of the tube wall can be combined in one term $R_s$.

Tube wall resistance for a smooth (non-enhanced) tube can be expressed as:

$$R_w = \frac{1}{h_w} = \frac{\ln (D_o/D_i) D_{ref}}{K \qquad 2} \qquad (3)$$

Where $D_{ref}$ is the reference diameter on which the overall U is based on. For a fluted tube the tube wall resistance is also based on the nominal diameter $D_n$, and equation can be written:

$$R_w = \frac{\ln (D_n/D_i) D_n}{2K} \qquad (4)$$

Therefore, the concept of resistances in series yields:

$$\frac{1}{U_n} = \frac{D_n/D_i}{h_f} + R_s + \frac{\ln(D_n/D_i)D_n}{2K_w} + \frac{1}{h_c} \qquad (5)$$

## IIB. WATERSIDE HEAT TRANSFER COEFFICIENTS

### 1. Smooth Internal Tube

For the smooth internal tube the McAdams correlation was used for determination of the heat transfer coefficient for cooling water.

$$\left[\frac{hD}{K_b}\right] = 0.023\left[\frac{GD}{\mu_b}\right]^{0.8}\left[\frac{\mu Cp}{k}\right]^{0.4} \qquad (6a)$$

or:

$$h_f = 0.023\ \frac{K_b}{D}\left[Re_D\right]^{0.8}\left[Pr\right]_b^{\ 0.4} \qquad (6b)$$

where:

1) all fluid properties are evaluated at the bulk fluid temperature;

2) $2300<Re_D<10^7$ where;

$Re_D$ = Reynolds number based upon hydraulic diameter

3) $0.5<Pr_b<120$ where;

$Pr_b$ = Prandtl number based on bulk temperature

### 2. Helical Internal Ridging

Internal tube augmentation was also investigated, with the application of integral multiple helix ridging [6] as shown in Figure 2.

Friction factor data was correlated by means of the following equation:

$$\sqrt{\frac{f}{8}} = \frac{1}{2.46\left[\ln r+(7/Re)^m\right]} \qquad (7)$$

For the roughness of the helical internal ridge both r and m are treated as variables, and are tied in with tube geometry. Both m and r vary with the dimensionless parameter e/l, where e is the ridge height and l is the lead of the ridge. In reference [6] WITHERS has made a distinction based on

the criterion p/d= 0.36 in correlating the friction behavior to the internal geometry. It has been proposed that a shift in flow behavior occurs at p/d = 0.36. For higher values of p/d a greater degree of swirling could occur, compared with cascading of flow that takes place if p/d is less than 0.36.

The heat transfer correlation equation developed from data for tubes of various configurations, and solved for the Stanton number becomes:

$$St = \frac{\sqrt{f/8}}{5.68(e/p)^{-1/8}\sqrt{Pr}\,[(e/d_i)Re\ f/8]^{0.136} + \gamma} \qquad (8a)$$

where: $$\gamma = -[2.5\ \ln(2e/d_i)+3.75] \qquad (8b)$$

Then, $h_f$ becomes:

$$h_f = \frac{K_c\ Re\ Pr\ St}{d_i} \qquad (9)$$

These equations are applicable to Reynolds number range 10,000-120,000, and Prandtl number range 4-10.

See Appendix-B for tube data, and friction factor characteristics of multiple-helix internal ridged tubes.

## II.C CONDENSATE HEAT TRANSFER COEFFICIENT

In reference-1, BARNES developed an equation for the average value of condensate heat transfer coefficient ($h_c$) for an externally fluted tube of length L. In this formulation $h_c$ depends upon physical properties of the condensing fluid geometric factors, and mass flow rate of condensate in the flutes.

$$\bar{h}_c = .6027\left[\frac{h_{fg}W_f}{L\Delta T}\right]^{.0074} \frac{a^{.2307}(Nu_o\Omega^{\frac{1}{4}})^{.9226}}{p} \left[\frac{K^3\rho\sigma h_{fg}g_e}{\mu\Delta T}\right]^{.2307} \qquad (10)$$

The non-dimensional group $Nu_o\Omega^{\frac{1}{4}}$ defined in Reference-1 is a function of flute amplitude-to-pitch (a/p) ratio. It should be noted in Equation-10 the parameter L is the length of the tube between condensate drain off plates.

The parameter $W_f$ is the flooding axial mass flow of condensate per flute (LBM/hr) which is also a function of physical properties of the fluid, and tube geometry. In Reference-4 PANCHEL and BELL defined the following flooding Reynolds number for condensate flow in the axial flutes:

$$Re_f = \frac{4Wf}{\mu X_L} \qquad (11)$$

and an additional non-dimensional group:

$$\lambda = \frac{4\rho^2}{\mu^2} \; g_c\left[\frac{\alpha_o}{X_L}\right]^4 \qquad (12)$$

where $\alpha_o = 2a$ in the case where the flute is flooded. Also, the following correlation was developed for $\lambda_f$ based on the

half-perimeter length ($X_L$) of the flute:

$$\lambda_f = 36(a/p)\exp(3.33\ a/p)\ Re_f \qquad (13)$$

Substituting equations (11 and 12) into equation (13) and solving for $W_f$ yields:

$$W_f = \frac{8}{9}\ g_c\ \frac{\rho^2}{\mu}\ a^3 p\ \exp\left[-3.33\left(\frac{a}{p}\right)\right] \qquad (14)$$

where $W_f$ is now based on full perimeter length of the flute. See Appendix-B for plot of $Nu_o\Omega^{\frac{1}{4}}$ versus a/p ratio.

## III. PROCEDURE DISCRIPTION FOR HEAT EXCHANGER SIZING

The following discussion outlines the basic methodology for condenser sizing used in this analysis. At this point in this discussion, it is assumed that basic condenser operational parameters have already been determined, i.e., condenser heat load cooling water flow rate, number of tubes, coolant temperature rise from inlet to outlet of condenser, tube size, etc.

1. In the first step, divide the total coolant temperature rise from condenser inlet to outlet into small increments of $\Delta T_s$ each, and assume constant fluid properties over each $\Delta T_s$ increment. If the coolant temperature rise for each increment is known, then the incremental heat transfer is also known.

2. The next step would be to assume initial values for L and $\Delta T$ ($T_{sat}$ - $T_{wall}$) in Equation (10).

3. Calculate overall $U_n$ Equation (5) for the increment after heat transfer coefficients are determined.

4. The section length is determined, then a heat balance is applied to the length increment to obtain new calculated values for $\Delta T$. These updated values for $\Delta T$ and L will serve as input into Equation (10) for the next iteration.

5. Repeat steps 2-4 until the calculated $\Delta T$, and L in step 4 equals the assumed $\Delta T$, and L in step 2.

6. In this manner one can march through the condenser summing the individual increment lengths with the final result being the total tube length required for the condenser.

Tube bundle diameter was then determined using acceptable values for pitch-to-diameter ratio, which was taken as 1.35 here. Tube spacing determined by the pitch-to-diameter ratio

is based on the tube outside diameter including the flutes. Calculated values for the steam lane dimensions along with tube bundle dimensions were used to determine the internal dimensions of the condenser shell.

The steam lane is the radial steam passage around the tube bundle, whose width is the radial distance between the outer tube bundle diameter and the condenser shell. A simple expression was established to determine the steam lane width using emperical data from baseline condensers. It is desired to size the steam lane width to limit the main steam lane entrance velocity to the below recommended values:

| Condenser Design Pressure, in. Hg | Recommended Maximum Main Steam Lane Entrance Velocity(ft/sec) | |
|---|---|---|
| 1 | 500 | |
| 2 | 400 | |
| 3 | 300 | (ref 9) |
| 4 | 250 | |
| 5 | 200 | |

This steam lane velocity can be expressed:

$$V = \frac{1}{3600} \frac{W}{\rho A} \qquad (15)$$

$V =$ steam velocity (ft/sec)

$W =$ steam load (lb/hr)

$A =$ flow area (ft$^2$)

$\rho =$ steam density (lb/ft$^3$)

Equation (16) can be rewritten:

$$V = \frac{1}{3600} \frac{W}{\rho L_T 2 L_W} \qquad (16)$$

$L_T =$ total tube length (ft)

$L_W =$ steam lane width (ft)

Since some of the steam enters directly into the tube bundle, W in the above equation should be reduced by an appropriate amount. The resulting equation that follows

agrees well with similar condenser designs:

$$V = \frac{.8784}{3600} \quad \frac{W}{\rho L_T 2 L_W} \qquad (17)$$

Therefore, up to this point in the procedure the following basic information has been determined:

1. Tube Length of condenser shell
2. Condensate drainage plate spacing
3. Tube bundle dimensions and volume
4. Condenser shell dimensions and volume

The remainder of the sizing procedure concerns the determination of component dimensions for the tubesheets, waterboxes, hotwell, etc. The required material, dimensions, arrangement, and construction of condenser components were determined in accordance with Reference (7), reflecting submarine design practice.

The final product of this procedure is preliminary condenser sizing and arrangement, and the following values for performance comparison: total condenser weight and volume, and the cooling water system pressure drop through the condenser.

# IV. COMPUTER MODELING

The basic procedure for heat exchanger sizing as outlined in the previous section was employed in the Vertical Condenser Sizing Program No. 1 (VERTCON-1). The VERTCON-1 program is a preliminary design tool with the following features:

1. The option of smooth internal tubes or internally enhanced tubes.
2. Design of single or double pass condensers.
3. Option of submarine, or surface ship condenser design.
4. Two basic configurations are available for double pass condensers:
   (a) Conventional return header design
   (b) "U-Tube" construction
5. Diagnostic messages used to warn when condenser drainage plate spacing is beyond specifications based on heat transfer performance and mechanical design.
6. Computation of recommended values for condenser drainage plate spacing and maximum main steam lane entrance velocity when they are not specified.

## Program Description

The program VERTCON-1 is described here, and diagrammatically in the simplified program flow chart (Figure 5). The program is based on the FORTRAN-77 standard, and is listed in Appendix-A. Included with the program listing are sample

data input files, and program output.

The program input consists of input data files, and interactive input entered at a terminal. The input data files consist of tube geometry information, and specification of condenser operating conditions. The design features are selected interactively at the terminal. See Appendix-A for further details on data input.

The program is set up for sizing just one condenser or a series of condensers. After inputing data the program prompts at the terminal for selection of design features. If condenser drainage plate spacing and maximum main steam lane entrance velocity are not specified the program calculates recommended values (Branch A and Iteration-A, Figure 5). The determination of these values are based on heat transfer performance and mechanical design to provide proper tube support. If these values are specified then the program checks if condenser drainage plate spacing is beyond recommended values (Branch B, Figure 5). If the values specified exceed those dictated by proper design, then diagnostic warning messages are printed at the terminal and in the program output. For an example of this see sample output condenser number 4 in Appendix A.

In Iteration B and C, the total effective tube length for the condenser is calculated by adding N tube sections of length equal to the drainage plate spacing until a proper heat balance is obtained. Iteration B adds up N-1 of these sections, and Iteration C calculates final section length, which may have a length less than the specified drainage plate spacing.

After total effective tube length is determined the condenser geometry, weight, volume and system pumping power are calculated for each condenser. The tabulated program output displays for each condenser the following information:

(1) All input data and design features selected.

(2) If applicable, diagnostic warning messages.

(3) Detailed weight breakdown of condenser components.

(4) Total condenser volume center of gravity and total weight in dry and wet conditions, and system pumping power.

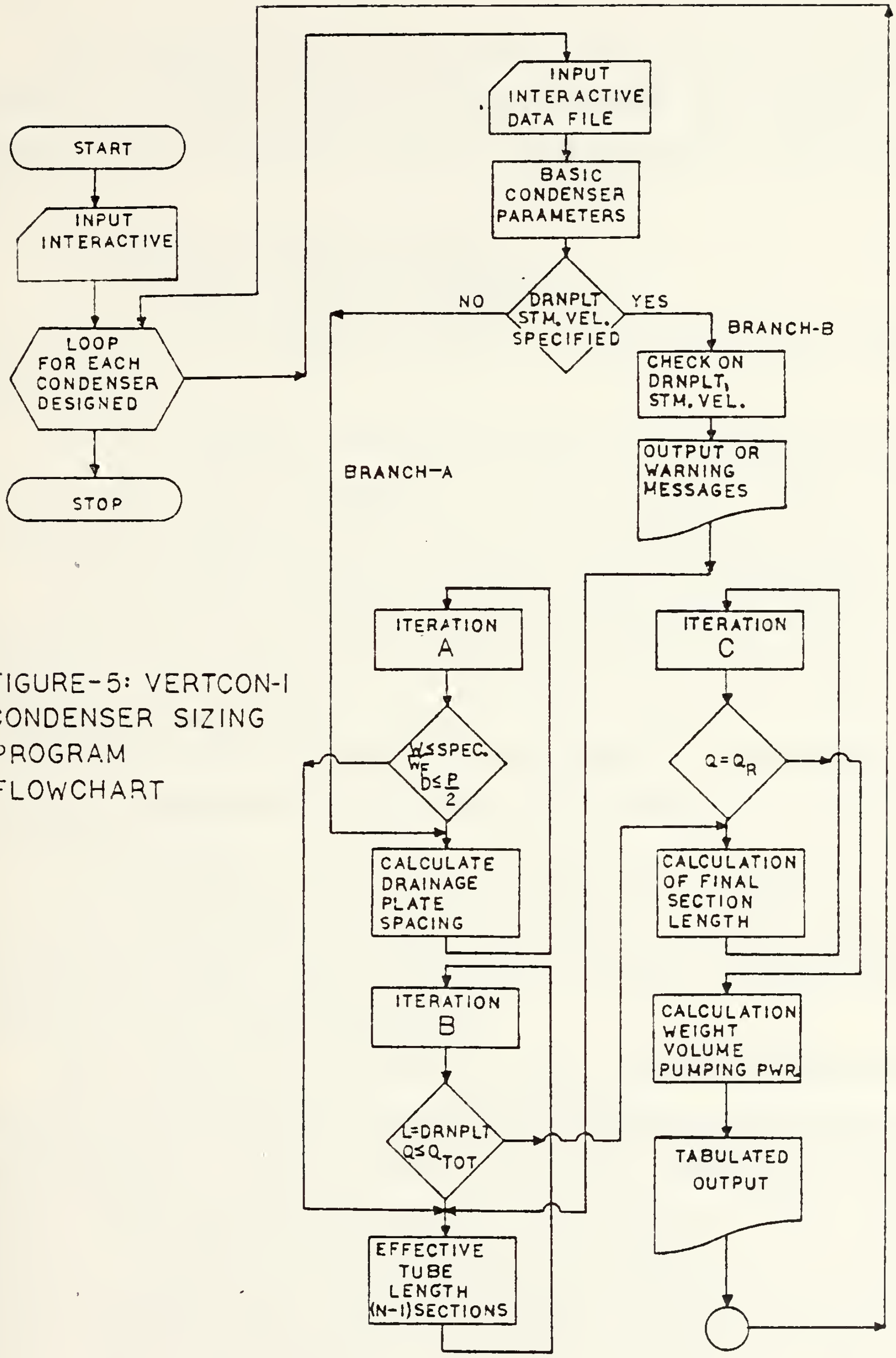



FIGURE-5: VERTCON-I CONDENSER SIZING PROGRAM FLOWCHART

## V. TUBE SELECTION

Three basic groups of tubes were used in the vertical condenser sizing routine and subsequent performance comparison with horizontal smooth and enhanced tubes, and they are:

1. smooth internal tube
2. mild internal enhancement
3. extreme internal enhancement

Also tubes were compared with the same degree of external enhancement.

For all tubes, the outer tube wall diameter was 5/8 in. with a tube wall thickness of 0.035 in. The base of the axial flutes are located at the outer wall diameter ($D_W$).

In selection candidates for internal enhancement various configurations were investigated from reference (6). The objective was to obtain enhancement configurations with the following properties:

1. Candidate tubes with acceptable improvement in internal enhancement with lower cooling water pressure drop characteristics.

2. Candidate tubes with significant improvement in internal enhancement. Due to the more extreme internal enhancement these tubes will have higher cooling water pressure drop characteristics.

The selection criteria for the external enhancement considered both heat transfer performance and tube strength requirements. For a fluted tube under internal pressure, stress concentrations occur at the bottom of the flute valley. A method was developed by NEUBER (8) to relate stress concentration factor to flute amplitude-to-pitch ratio. With information obtained by applying this method, a criteria can be established for selection of a maximum

a/p ratio. The effect of a/p ratio on tube strength will only be significant at deep operating depths for submarine condensers, and it will not be a strong influencing factor in surface ship condenser designs.

The condensate heat transfer coefficient $h_c$ is also a strong function of a/p ratio, and it generally increases in the range of a/p up to approximately 0.35. Work is ongoing in this area to determine selection criteria for optimal a/p ratio.

The desire to keep the a/p ratio at higher values for improved heat transfer performance is opposed by the strength requirements that place a limit on the maximum value for this ratio. The result of balancing these two opposing requirements resulted in the tube selections as shown below:

Table 1. Enhanced Vertical Tube Data

Tube Wall Diameter = 5/8 in.
Tube Wall Thickness = 0.035 in.

Cooling Water Enhancement

| Version Number | m | r | e (in.) | p (in.) | e/di | e/Pi | f* |
|---|---|---|---|---|---|---|---|
| 1 | .58 | .0075 | .0240 | .0949 | .04324 | .2529 | .072 |
| 2 | .59 | .00197 | .0204 | .1910 | .03676 | .1070 | .043 |
| 3 | .762 | 0 | .0125 | .4750 | .02252 | .0263 | .030 |

*Friction Factor @ Re = $4.0 \times 10^4$

Steam Side Enhancement

| Version Number | $A_o$ (in.) | $P_o$ (in.) | $a_o/P_o$ | $Nu_o \Omega^{\frac{1}{4}}$ | Number of Flutes |
|---|---|---|---|---|---|
| 1 | .01 | .0409 | .245 | 4.54 | 48 |
| 2 | .015 | .0874 | .1749 | 4.40 | 24 |
| 3 | .008 | .0336 | .2384 | 4.487 | 60 |

## VI. COMPARISON OF HEAT TRANSFER RESISTANCES

A comparison of steamside, tube wall, and waterside resistances for typical vertical, horizontal smooth and enhanced tubes is depicted in Figure (6). All three tubes are operating at the same flow velocities and condensing conditions. Both the horizontal and vertical enhanced tubes are compared with the same degree of internal enhancement, and thus the cooling water pressure drop experienced by these two enhanced tubes are also approximately equal. Also, the cooling water pressure drop for the enhanced tubes is greater than for the smooth horizontal tube.

The numbers on the bar graph in Figure (6) for the horizontal and vertical enhanced tubes indicate the overall reduction in thermal resistance as compared with the horizontal smooth case. For the horizontal enhanced case the 26 percent thermal resistance reduction corresponds to a single helix ridged tube, and the 35 percent thermal resistance reduction corresponds to a single helix ridged tube where additional external enhancement surfaces (grooves) have been added. As demonstrated, the horizontal enhanced tube provides for significant thermal resistance reduction on the waterside, and reduction to a limited extent on the steam side.

As compared with both horizontal smooth and enhanced tube the vertical tube provides for significant reduction in steamside thermal resistance. This decrease in steamside resistance doesn't occur without some cost, which in this case turns out to be an increase in tube wall resistance. The increase in tube wall resistance occurs because the axial flutes are built up on the tube at the wall diameter thus increasing the length of the heat transfer path through tube wall material. This increased tube wall resistance is reflected in the formulation

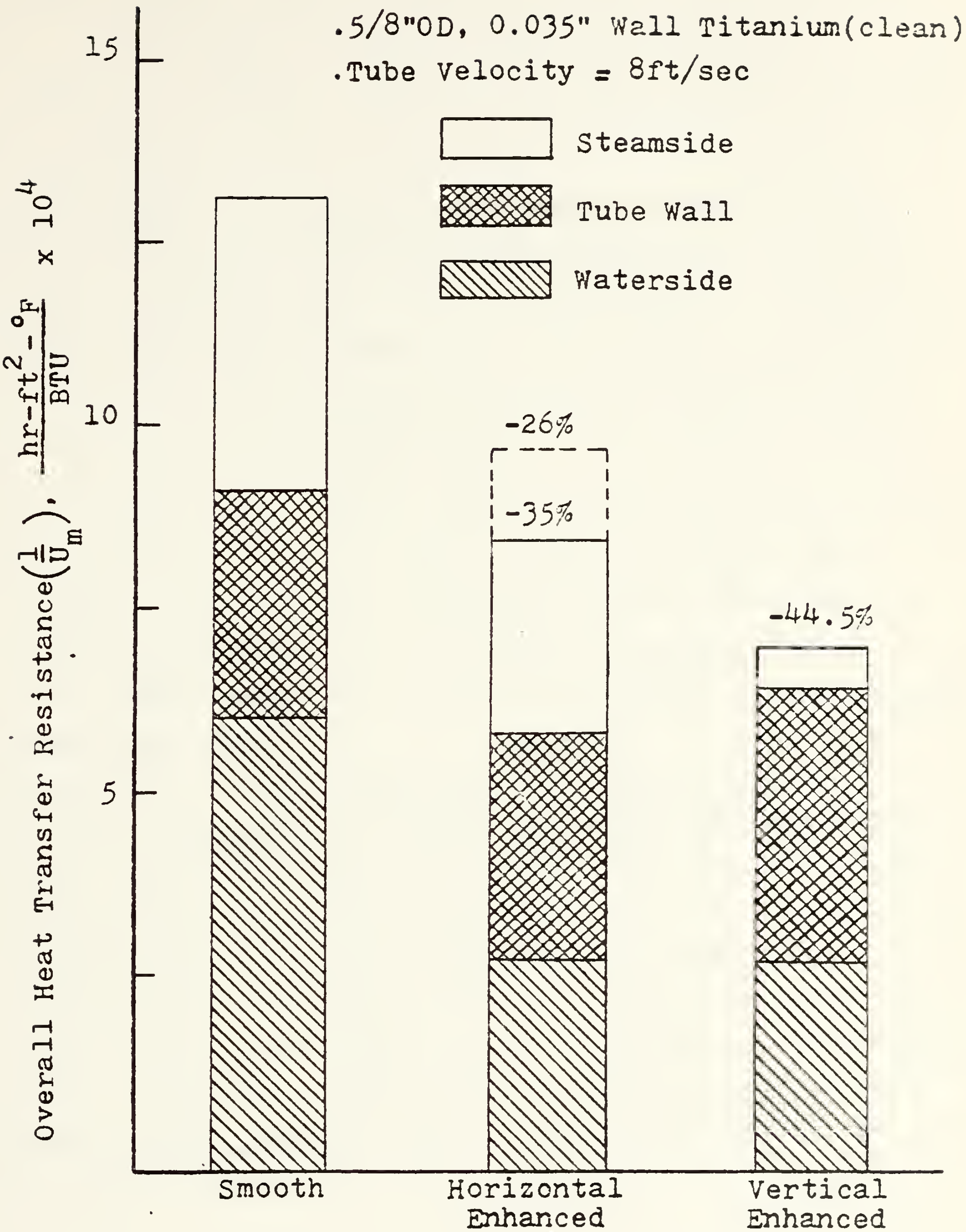



FIGURE 6

Comparison of Heat Transfer Resistance

of the heat transfer coefficient for the tube wall in equation (4). As horizontal enhanced tubes are modified to provide improved steamside enhancement by more severe external grooving similar problems with increased tube wall resistance will be encountered.

This increase in tube wall resistance can be minimized by reducing the value of the flute amplitude (a). In order to maintain the amplitude-to-pitch (a/p) ratio at a specified value, correspondingly the value of p will have to be reduced. The overall effect of reducing both a and p to minimize the increase in tube wall resistance will be to produce a tube with a large number of smaller flutes. As discussed before, this sort of trend will improve heat transfer performance up to a point where the flutes become so small that they are easily flooded. This flooding problem could be partially offset by also decreasing the condensate drainage plate spacing, but again there is a limit to how small the drainage plate spacing can be.

The material for all three tubes in figure (6) is titanium. The comparative increase in tube wall resistance for vertical tubes will not be as significant for materials possessing greater thermal conductively, such as CuNi or Al.

As demonstrated, vertical enhanced tube can cause a noteworthy reduction in overall thermal resistance by providing significant reductions in both waterside and steamside resistance. It should be noted that to insure minimum tube wall resistance care must be taken in the selection of the axial flute geometry.

## VII. RESULTS OF PERFORMANCE COMPARISON

The following series of four figures demonstrate the comparison of condenser performance between various vertical tube condenser configurations, and the horizontal enhanced and baseline horizontal smooth tube condensers. Enhanced and baseline condenser data is displayed in Table 2. The condensers are characterized as being either High Condenser Absolute Pressure (HCAP) or Low Condenser Absolute Pressure (LCAP) designs.

The LCAP and HCAP are typical submarine condenser designs, and Code 2721, David Taylor Naval R & D Center was the source of data for the horizontal enhanced and the baseline horizontal smooth tube condensers. This data is based on the results out of the ORCON-2 program from the Naval Postgraduate School, Monterey. For the vertical tube condensers equivalent procedures were used in determining the weight and dimensions of condenser components, to ensure a valid comparison of condenser performance.

The performance comparison is presented by plotting total condenser weight ratio ($W_e/W_{bl}$) versus the pumping power ratio ($P_e/P_{bl}$). The ratio $W_e/W_{bl}$) is the weight of the enhanced tube condensers divided by the weight of the baseline horizontal smooth tube condenser. The condenser weight is the total wet weight of the condenser and its components, excluding external pumps and piping. The ratio $P_e/P_{bl}$ is the pumping power of the enhanced tube condensers divided by that of the baseline horizontal smooth tube condenser. The pumping power is a combined value for both the condenser and the condenser seawater circulating system. A plot of total condenser volume ratio ($V_e/V_{bl}$) versus $P_e/P_{bl}$ ratio was also made, where condenser volume is the total box volume of condenser and components excluding external piping and pumps.

TABLE 2 - ENHANCED AND BASELINE CONDENSER DATA

| | HCAP Enhanced Vertical | LCAP Enhanced Vertical | HCAP/LCAP Enhanced Horizontal & Baseline Smooth Tube |
|---|---|---|---|
| Tube spacing/ diameter | 1.35 | 1.35 | 1.35 |
| Tube OD(in.) | 0.625 | 0.625 | 0.625 |
| Tube Wall Thickness(in.) | 0.035 | 0.035 | 0.035 |
| Tube Material | Ti | Ti | Ti |
| Tube Wall Conductivity (BTU/hr ft °F) | 9.5 | 9.5 | 9.5 |
| Cooling Water Velocity (ft/sec) | 8.0-14.0 | 8.0-14.0 | - |
| Cooling Water Inlet Temp.(°F) | 66.1 | 66.1 | 66.1 |
| Fouling Resistance ($hrft^2 °F/BTU$) | 0.00033 | 0.00033 | 0.00033 |
| Steam Inlet Saturation Temperature(°F) | 143.89 | 112.0 | 143.89/112.0 |
| Cooling Water Outlet Temp(°F) | 123.46 | 98.28 | 123.46/98.28 |
| S.W. Flow (GPM) | 7900 | 13,500 | 7900/13,500 |
| Condenser Operating Pressure (in.Hg) | 6.5 | 1.5 | 6.5/1.5 |

BASELINE HORIZONTAL SMOOTH TUBE CONDENSER DATA

| | HCAP | LCAP |
|---|---|---|
| S. W. Flow (GPM) | 7900 | 13500 |
| Cooling Water Velocity (ft/sec) | 6.36 | 7.67 |
| Overall Width (ft) | 6.25 | 8.06 |
| Overall Length (ft) | 25.3 | 28.8 |
| Total Wet Weight (lbs) | 110,303 | 143,369 |
| Overall Head Loss (ft-$H_2O$) | 15.4 | 24.3 |
| Box Volume ($ft^3$) | 1226 | 2091 |
| System Pumping Power (HP) | 190 | 277 |

## HCAP - ENHANCED-VERTICAL TUBE CONDENSER PERFORMANCE

Performance comparison for the HCAP condenser design on the basis of $W_e/W_{bl}$ versus $P_e/P_{bl}$ is plotted in Figure 7 and $V_e/V_{bl}$ is plotted in Figure 8. Three different internal enhancement configurations are employed all with the same steam side enhancement (NR-1).

### Curve-3

Data for smooth internal vertical tubes are plotted in curve-3. The range of condenser parameters going from left to right on the curve are:

| | |
|---|---|
| Cooling water velocity(ft/sec) | 6.36 - 8.0 |
| Total condenser length(ft) | 24 - 26 |
| Effective tube length(ft) | 14.0 - 16.3 |
| Condenser diameter(ft) | 6.8 - 5.9 |
| Box volume($ft^3$) | 1178. - 978 |
| Condenser weight(lbs) | 86,763 - 71,900 |
| Pumping power(HP) | 184. - 210 |

The effective tube length is only the tube length required to transfer the specified heat load, and it doesn't include the extra tube length required for double tube sheet construction. Traveling along the curve from left-to-right the following trends can be observed:

1. cooling water velocity increases
2. total condenser length and effective tube length both increase
3. condenser diameter, box volume, and weight all decrease
4. pumping power increases

These same general trends can be observed in all the plotted data for both the HCAP and LCAP designs. As shown above, condenser diameter decreases as the condenser length and flow velocity increases with a corresponding reduction in both weight and volume. This demonstrates that a small condenser diameter and long length produces a greater weight and volume reduction. In practice however, a pumping power or practical condenser length limit will dictate an acceptable range for a design on this curve. This indicates that a condenser with in the greatest weight and volume reduction will also possess larger values for condenser length and pumping power.

This methodology can be applied to nonenhanced condenser designs, but the length limit will be much more restrictive than for enhanced condensers, thus restricting the degree of improvement.

This data also demonstrates that condenser weight and volume is a strong function of condenser diameter. Condenser box volume is a function of diameter squared, thus reducing diameter will have a greater effect in reducing volume than changing condenser length. Also, the largest weight condenser conponents have dimensions based on diameter, such as waterboxes, tubesheets, tube support plate and condenser shell. Therefore, reducing condenser diameter will have a significant effect in reducing weight.

Curve-3 also demonstrates that significant weight and volume reduction is possible with just steamside enhancement. If the application of internally enhanced tubes becomes restricted due to excessive rates of fouling, then vertical tube condensers will still be able to provide an enhanced condenser alternative with smooth internal tubes.

Curve-2

Data in Curve-2 are for condensers with doubly enhanced vertical tubes with mild degree of internal enhancement (internal enhancement NR-2). The range of condenser parameters going from the left to right on the curve are:

| | |
|---|---|
| Cooling Water velocity(ft/sec) | 8.0 - 10.0 |
| Total condenser length(ft) | 23 - 25 |
| Effective tube length(ft) | 12.7 - 15.3 |
| Condenser diameter(ft) | 6.5 - 5.6 |
| Box volume($ft^3$) | 1042 - 852 |
| Condenser weight(lbs) | 75,522 - 62,468 |
| Pumping power(HP) | 215 - 307 |

The condenser designs located on this curve show an additional reduction in weight and volume over the smooth internal vertical tube configuration. But along with the increased performance there is an increase in pumping power. Overall, the Curve-2 condensers provide the best design alternatives for the following reasons:

1. The mild internal enhancement coupled with steamside enhancement provides significant weight and volume reduction.
2. Condenser lengths fall within a reasonable range.
3. The mild enhancement may prove to produce less fouling problems than other more severely enhanced tubes.

Curve-1

Data in Curve-1 depict condenser designs with doubly enhanced vertical tubes with extreme internal enhancement (internal enhancement NR-1). The range of condenser parameters going from left to right on the curve are:

| | |
|---|---|
| Cooling water velocity(ft/sec) | 8.0 - 12.0 |
| Total condenser length(ft) | 22 - 26 |
| Effective tube length(ft) | 11.8 - 16.6 |
| Condenser diameter(ft) | 6.7 - 5.1 |
| Box volume($ft^3$) | 1076 - 747 |
| Condenser weight(lbs) | 78,620 - 54,721 |
| Pumping power(HP) | 280 - 533 |

The condensers with the greatest weight and volume savings are the designs on Curve-1. These same designs also have the largest values for pumping power.

As demonstrated by the HCAP designs in Figures 7 and 8, the vertical enhanced tube condensers provide a significant performance improvement compared with both enhanced horizontal and conventional horizontal smooth tube condensers. Vertical enhanced tube condensers accomplish this with a great degree of flexibility, due to the various combinations of internal and external enhancement that can be employed.

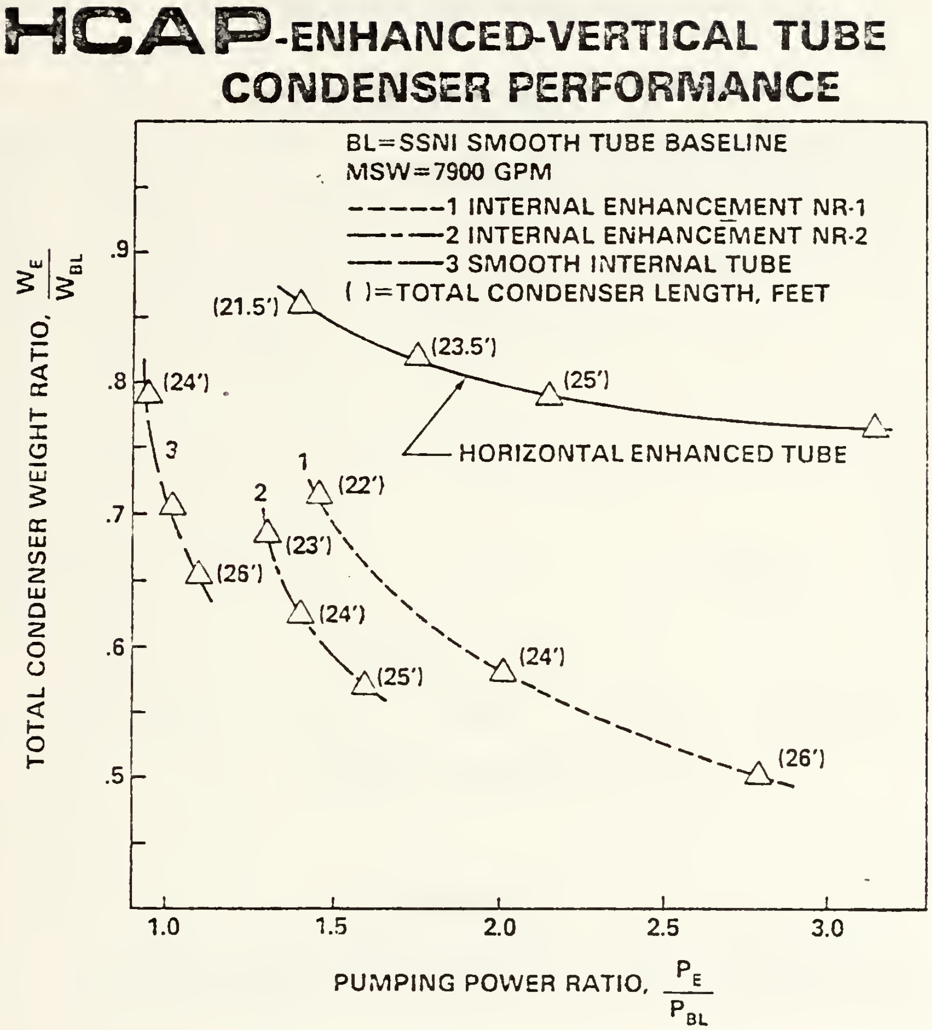



FIGURE 7

HCAP - Enhanced Vertical Tube Condenser Performance - Condenser Weight Comparison

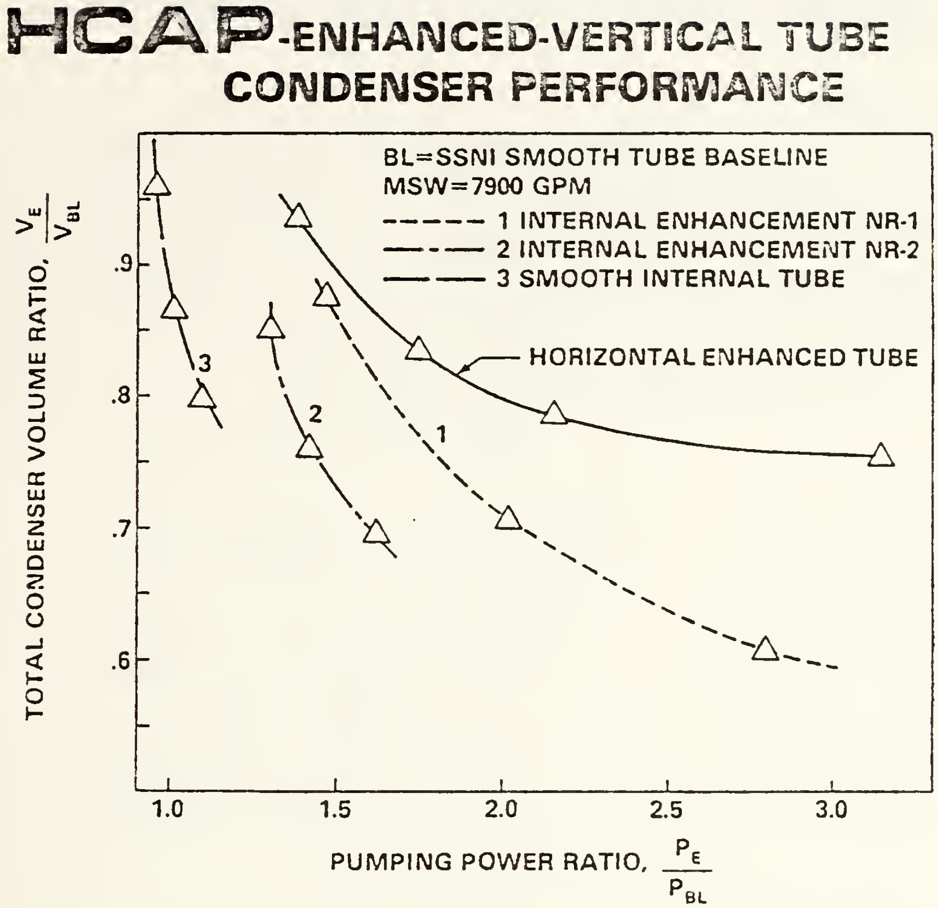



FIGURE 8

HCAP - Enhanced Vertical Tube Condenser Performance - Condenser Volume Comparison

## LCAP-ENHANCED-VERTICAL TUBE CONDENSER PERFORMANCE

Performance comparison for the LCAP condenser design on the basis of $W_e/W_{bl}$ versus $P_e/P_{bl}$ is plotted in Figure 9, and $V_e/V_{bl}$ is plotted in Figure 10. The vertical condenser designs depicted in this data are for only one tube configuration: internal enhancement NR-2, and external enhancement NR-3. The range of condenser parameters going from left to right on the curve are:

| | |
|---|---|
| Cooling water velocity (ft/sec) | 10.5 - 12.0 |
| Total condenser length (ft) | 26 - 27.5 |
| Effective tube length (ft) | 14.6 - 16.2 |
| Condenser diameter (ft) | 8.95 - 8.15 |
| Box volume ($ft^3$) | 2227 - 1920 |
| Condenser weight (lbs) | 130,847 - 112,614 |
| Pumping power (HP) | 443 - 533 |

Only one tube configuration was plotted due to design length limitations. With larger allowable condenser lengths additional weight and volume reductions could have been obtained with less internal enhancement, and at a lower pumping power. With greater degree of internal enhancement the pumping power limit was quickly reached.

At lower condenser operating pressures as in the LCAP design, weight and volume is a strong function of condenser diameter which is driven by steam lane volume. Steam lane volume is directly proportional to specific volume of steam and inversely proportional to condenser length. The magnitude of specific volume for steam is over twice the value in the LCAP design as compared with the HCAP design.

If condenser length is reduced by application of enhanced tubes while keeping other condenser parameters constant,

condenser weight and volume could actually be increased. Since steam lane volume is inversely proportional to condenser length, reducing condenser length with the same steam mass flow rate and tube bundle diameter will cause a significant increase in condenser diameter. This increase in condenser diameter will strongly influence an increase in weight and volume.

Condenser weight and volume can be reduced by boosting flow velocity, which causes both an increase in condenser length and a decrease in condenser diameter due to reduced tube bundle diameter (less number of tubes and reduced steam lane width). There is a limit to how far flow velocity can be increased, because the pumping power limit is quickly reached.

The effect of steam specific volume on condenser sizing has a greater effect on condenser designs with lower operating pressures such as the LCAP design that has been described above. Since the vertical condenser design is length limited, the amount of condenser performance improvement that can be achieved at lower condenser pressures is also limited. This trend is evident in the data of Figures 9 and 10. To illustrate low the vertical condenser design is length limited examine the data point for condenser length equal to 27.5 feet. The effective tube length is 16.2 feet, while the remaining 11.3 feet of condenser length is made up of waterbox header depth and hotwell depth. Relocation of the hotwell could provide several extra feet for increased effective tube length. With certain condenser conditions a one foot increase in condenser length could provide as much as 10 percent additional condenser performance improvement. Thus at lower condenser operating pressures, the vertical enhanced tube condenser performance is sensitive to allowable condenser length.

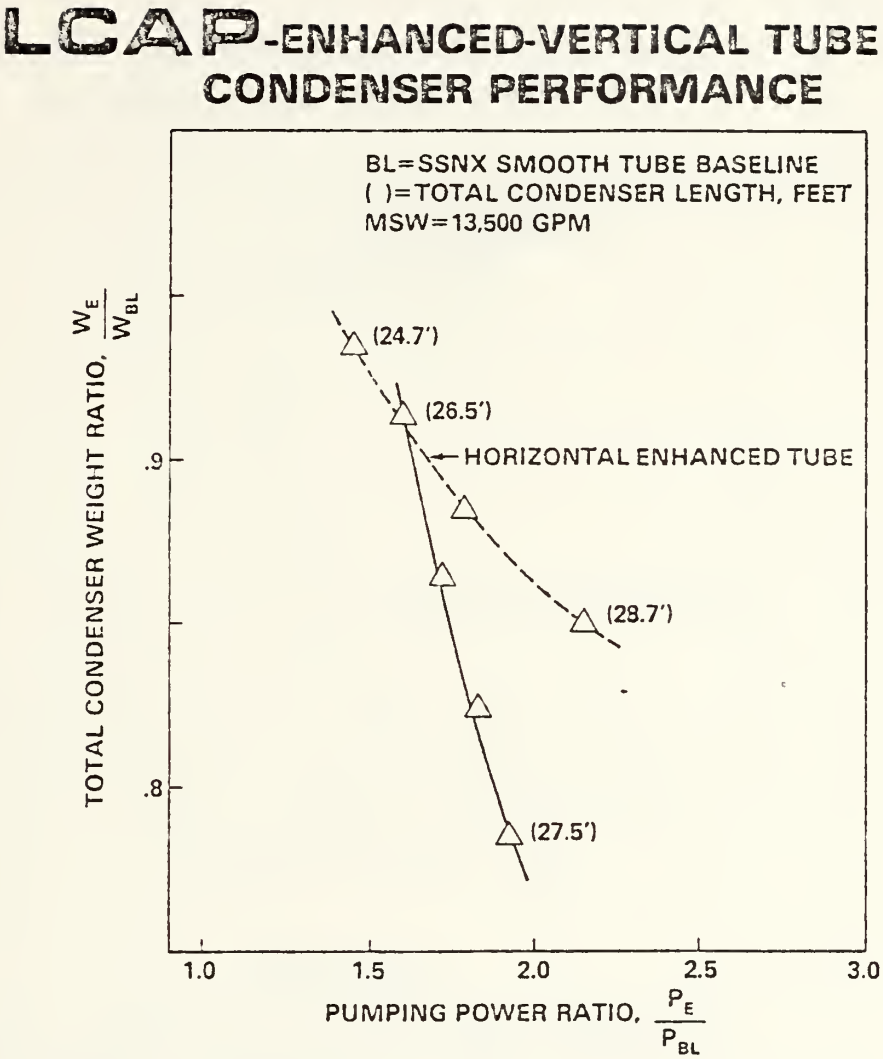



FIGURE 9

LCAP - Enhanced Vertical Tube Condenser Performance - Condenser Weight Comparison

# LCAP-ENHANCED-VERTICAL TUBE CONDENSER PERFORMANCE

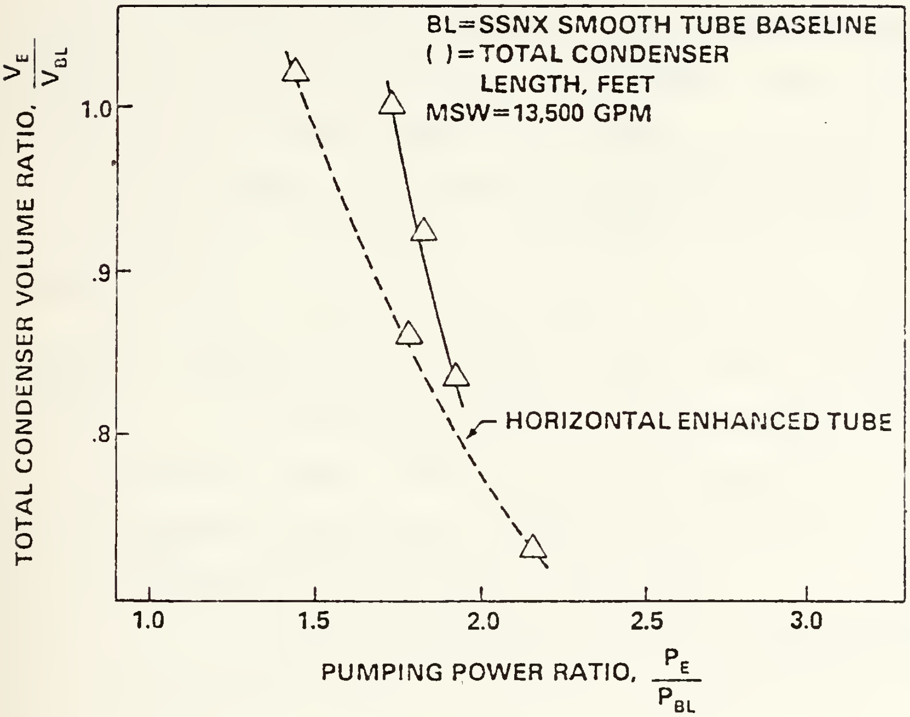



FIGURE 10

LCAP - Enhanced Vertical Tube Condenser Performance - Condenser Volume Comparison

## VIII. CONCLUSIONS

This analysis has clearly demonstrated that significant reduction in condenser weight and volume can be obtained for certain applications using enhanced vertical tube condensers. Vertical tube condenser performance also compares well with enhanced horizontal tube condensers. The following conclusions can also be drawn from this report:

1. Larger comparative weight and volume reductions can be obtained with designs at higher condenser operating pressures.

2. The proper methodology for producing enhanced condenser designs with minimum weight and volume is to:

   a. first determine proper value for average overall heat transfer coefficient (U) within pumping power and dimension limits to produce minimum weight and volume; and

   b. match internal and external enhancement configurations to obtain the specified value for U.

3. Vertical tubes can provide great flexibility due to the different combinations of internal and external enhancement configurations. It is possible to "fine tune" the overall heat transfer coefficient to obtain the optimal value of U. Condensate drainage plate spacing can also be used to adjust the value of condensate heat transfer coefficient.

4. Significant weight and volume reductions can be obtained at lower pumping powers with mild internal enhancement or with smooth internal tubes.

5. The degree of enhancement should be matched for each particular case. The greatest degree of enhancement may not always be the optimum enhancement with design limitations to produce minimum condenser weight and volume.

6. Vertical tube condenser design is sensitive to effective tube length, and it is restricted by the maximum allowable condenser length.

7. Alternate hotwell designs need to be investigated. Relocating the hotwell from directly underneath the inlet/ outlet cooling water header (Fig. 4) could serve to reduce total condenser length.

Further analytic investigations and testing should continue with enhanced vertical tube condensers. The vertical condenser could have other important applications other than propulsion steam condensers. This technology can be applied to many of the shipboard two-phase heat exchangers, whether the condensing fluid is steam, refrigerant, or other medium.

## APPENDIX A

### DATA INPUT

See Table-3 for a sample input data file

### INPUT DATA FILE

1. Assigned Logical Operator - 20 to specify input device
2. See lines 47-52 of program listing for explanation
3. Free Format
4. Six lines of data for each condenser
5. Variable names starting with I, J, K, L, M, or N are integer variables and others are real variables, except for specific cases as noted

### LINE 1

Enter: SUB or SUR

SUB - Submarine Condenser Design

SUR - Surface Ship Condenser Design

### LINE 2

Enter: NTYPE, DI, DW, AE AEPE, EI, PI

NTYPE - Tube Type: a. for smooth internal tube
b. for doubly enhanced tubes

DI - Internal Tube Diameter (in.)

DW - Tube Wall Diameter (in.)

AE - External Flute Amplitude (in.)

AEPE - Amplitude-To-Pitch Ratio of External Ridging

EI - Internal Helix Ridge Height (in.)

PI - Pitch of Internal Helix (in.)

Note: For smooth internal tube (NTYPE-1) enter ØØ for EI, PI. Values of EI and PI are obtained from reference 6, and values used in this study are shown in Table-1.

TABLE 3

```
  * * SAMPLE INPUT FILE * *
SUB
2,.555,.625,.01,.245,.024,.0949
48.,4.54,.58,.0075,9.6,.00033
66.1,143.89,.85,257000.,6.5,2.
6.,.625,.5625,.5625,2.64,488.
558.,488.,488.,488.,558.,282.,1.56E7,.56,488.
SUB
2,.555,.625,.01,.245,.024,.0949
48.,4.54,.58,.0075,9.6,.00033
66.1,143.89,.85,257000.,6.5,2.
6.,.625,.5625,.5625,2.64,488.
558.,488.,488.,488.,558.,282.,1.56E7,.56,488.
SUB
2,.555,.625,.01,.245,.024,.0949
48.,4.54,.58,.0075,9.6,.00033
66.1,143.89,.85,257000.,6.5,2.
6.,.625,.5625,.5625,2.64,488.
558.,488.,488.,488.,558.,282.,1.56E7,.56,488.
SUB
2,.555,.625,.01,.245,.024,.0949
48.,4.54,.58,.0075,9.6,.00033
66.1,143.89,.85,257000.,6.5,2.
6.,.625,.5625,.5625,2.64,488.
558.,488.,488.,488.,558.,282.,1.56E7,.56,488.
SUR
2,.555,.625,.01,.245,.024,.0949
48.,4.54,.58,.0075,9.6,.00033
66.1,143.89,.85,257000.,6.5,2.
0.,.625,.5625,.5625,1.5,488.
0.,488.,488.,488.,558.,282.,1.56E7,.56,488.
SUR
1,.555,.625,.01,.245,0.,0.
48.,4.54,0.,0.,9.6,.00033
66.1,143.89,.85,257000.,6.5,2.
0.,.625,.5625,.5625,1.5,488.
0.,488.,488.,488.,558.,282.,1.56E7,.56,488.
```

LINE 3

Enter: FLUTE, FE, MI, RI, KW, RSCALE

- FLUTE - Number of Flutes Per Tube
- FE - Operand for Condensing Heat Transfer Coefficient
- MI - Operand for Friction Factor, Helical Internal Ridging (Real Variable)
- RI - Operand for Friction Factor, Helical Internal Ridging
- KW - Conductivity of Tube Wall (BTU/hr-ft-°F)
- RSCALE - Fouling Resistance (hr-ft-°F/BTU)

Note: MI is a real variable.

LINE 4

Enter: TCI, TSAT, QSTM, STMLD, PSAT, TIS

- TCI - Inlet Coolant Temperature (°F)
- TSAT - Steam Inlet Saturation Temperature (°F)
- QSTM - Quality of Inlet Steam to Condenser
- STMLD - Steam Condenser (lb/hr)
- PSAT - Condenser Operating Pressure (in. Hg) Absolute
- TIS - Thickness of Internal Tube Sheet (in.)

LINE 5

Enter: TOTS, TSHL, TTSP, THW, THDR, ITSD

- TOTS - Thickness Outer Tube Sheet (in.)
- TSHL - Thickness of Condenser Shell (in.)
- TTSP - Thickness Tube Support Plate (in.)
- THW - Thickness of Hot Well Plate (in.)
- THDR - Thickness of Headers (in.)
- ITSD - Internal Tube Sheet Density (lb./Ft$^3$) (Real Variable)

Note: ITSD is a real variable.

LINE 6

Enter: OTSD, SHLD, TSPD, HWD, HDRD, TBD, ETUBE, TCOVER,COVERD

- OTSD - Outer Tube Sheet Density (lb/ft$^3$)
- SHLD - Shell Density (lb/ft$^3$)

TSPD - Tube Support Plate Density ($lb/ft^3$)
HWD - Hot Well Plate Density ($lb/ft^3$)
HDRD - Header Density ($lb/ft^3$)
TBD - Tube Bundle Density ($lb/ft^3$)
ETUBE - Modulus of Elasticity of Tube Material ($lb/in^2$)
TCOVER - Thickness of U-Tube Header Cover (in.)
COVERD - Density of U-Tube Header Cover (in.)

INTERACTIVE DATA INPUT

Enter data at the terminal as follows:

° When the program commences, the following message appears on the terminal screen:

** WELCOME TO PROGRAM: VERTCON, VERSION-1 **
ENTER PROGRAM RUN NUMBER

Enter Run number, which can be an integer from 0-to-999.

Next at the terminal appears:

ENTER THE NUMBER OF CONDENSERS TO BE SIZED

Enter The number of condensers to be sized, which is also an integer from 1-to-999.

Next at the terminal appears:

ENTER "1" FOR SINGLE PASS, OR "2" FOR DOUBLE PASS CONDENSER

Enter the proper integer number 1 or 2 for single or double pass condenser.

Next at the terminal appears:

```
TWO BASIC CONFIGURATIONS ARE AVAILABLE FOR DOUBLE
PASS CONDENSERS:

(1)  CONVENTIONAL RETURN HEADER DESIGN
(2)  "U-TUBE" CONSTRUCTION

ENTER 1 OR 2 FOR CONFIGURATION SELECTION
```

Enter the proper integer number 1 or 2 for configuration selection.

Next at the terminal appears:

```
ENTER:  (1) COOLANT FLOW RATE (GPM)
        (2) COOLANT FLOW VELOCITY (FT/SEC)
        (3) FRICTION FACTOR FOR INTERNAL TUBE
```

Enter all three numbers (real numbers) on the same line in free format.

Note: Friction factors for internally enhanced tubes can be obtained from reference (6) and values used in this study are shown in Table-1.

Next at the terminal appears:

```
DO YOU WANT TO SPECIFY MAXIMUM MAIN STEAM LANE
ENTRANCE VELOCITY (FT/SEC)?

YES OR NO
```

Enter YES or Y, NO or N for proper choice.

If YES is entered, next at the terminal appears:

ENTER MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC)

Enter this as a real number.

If NO was entered, <u>next at the terminal secrrn appears</u>:

PROGRAM WILL SELECT RECOMMENDED MAXIMUM MAIN
STEAM LANE ENTRANCE VELOCITY USING CONDENSER
OPERATING PRESSURE AS SELECTION CRITERIA.

<u>Next at the terminal appears</u>:

DO YOU WANT TO SPECIFY CONDENSATE DRAINAGE
PLATE SPACING (FT.)?

YES OR NO

Enter YES or Y, NO or N for proper choice.

If YES, is entered, <u>next at the terminal appears</u>:

ENTER CONDENSATE DRAINAGE PLATE SPACING (FT.)

Enter this as a real number.

If NO was entered, <u>next at the terminal appears</u>:

PROGRAM WILL SELECT RECOMMENDED CONDENSATE
DRAINAGE PLATE SPACING

Note: At this point, entry of Interactive Data is completed for the first condenser. The program now completes the sizing for the first condenser and will prompt the terminal for further interactive data for any additional condensers.

DATA OUTPUT

See Table-4 for a sample output data file. This output data file demonstrates the design features as follows:

CONDENSER NUMBER (1)

1. Condenser Type: Submarine design
2. Double pass condenser with conventional return header design
3. Doubly enhanced tubes

CONDENSER NUMBER (2)

1. Condenser Type: Submarine design
2. Double pass condenser with "U-tube" construction
3. Doubly enhanced tubes

CONDENSER NUMBER (3)

1. Condenser Type: Submarine design
2. Single pass condenser
3. Doubly enhanced tubes

CONDENSER NUMBER (4)

1. Demonstration of program diagnostic warning messages

CONDENSER NUMBER (5)

1. Condenser Type: Surface ship design
2. Double pass condenser with conventional return header design
3. Doubly enhanced tubes

CONDENSER NUMBER (6)

1. Condenser Type: Surface ship design
2. Single pass condenser
3. Smooth internal tubes

TABLE 4

******** SAMPLE OUTPUT FILE ********

** PROGRAM: VERTCON,VERSION-1 **

PROGRAM RUN NUMBER( 1)

**** VARIABLE LIST ****

AE= EXTERNAL FLUTE AMPLITUDE (IN.)
AEPE= AMPLITUDE-TO-PITCH RATIO OF EXTERNAL RIDGING
ATB= AREA OF TUBE BUNDLE (FT**2)
ATS= AREA OF TUBE SHEET (FT**2)
DI= INTERNAL TUBE DIAMETER (IN.)
DPTHDR= HEADER DEPTH (FT.)
DRYCG= HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM AT DRY WEIGHT (FT.)
DTB= DIAMETER OF TUBE BUNDLE (FT.)
DTS= DIAMETER OF TUBE SHEET(FT.)
DW= TUBE WALL DIAMETER (IN.)
EI= OPERAND FOR FRICTION FACTOR HELICAL INTERNAL RIDGING
FE= OPERAND FOR CONDENSING HEAT TRANSFER COEFFICIENT
KW= CONDUCTIVITY OF TUBE WALL (BTU/HR-FT-DEG.F)
LANE= STEAM LANE BREADTH (FT.)
LHW= LENGTH OF HOTWELL (FT.)
LTOT= TOTAL TUBE LENGTH (FT.)
TCI= INLET COOLANT TEMP. (DEG.F)
MI= OPERAND FOR FRICTION FACTOR; HELICAL INTERNAL RIDGING
NTYPE= TUBE TYPE: 1-FOR SMOOTH INTERNAL TUBE; AND 2-FOR DOUBLY ENHANCED TUBES
PI= PITCH OF INTERNAL HELIX(IN.)
PSAT= CONDENSER OPERATING PRESSURE (IN.HG)ABSOLUTE
QSTM= QUALITY OF INLET STEAM TO CONDENSER
RI= OPERAND FOR FRICTION FACTOR, HELICAL INTERNAL RIDGING
RSCALE= FOULING RESISTANCE (HR-FT**2-DEG.F/BTU)
STMLD= STEAM LOAD (LBM/HR)
TSAT= SATURATION TEMP. (DEG.F)
VTB= VOLUME OF TUBE BUNDLE (FT**3)
WEXP= EXPANSION JOINT WEIGHT (LB.)
WMISC= WEIGHT OF MISCELLANEOUS COMPONENTS (LB.)
WETCG= HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM AT WET WEIGHT (FT.)

```
                   *****************************
                       CONDENSER NUMBER(  1)
                   *****************************
. . CONDENSER TYPE: SUBMARINE DESIGN
. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HEADER DESIGN.
. . DOUBLY ENHANCED TUBES . .
                          **** DATA INPUT ****
. . FLOW RATE:  7900. (GPM)  . . FLOW VELOCITY: 10.00 (FT/SEC) . .
DI = 0.555    DW = 0.625    AE = 0.010    AEPE = 0.2450    NTYPE =  2
EI = 0.0240   PI = 0.0949   FE =  4.54    MI = 0.580       RI = 0.0075
KW =   9.60   TCI =  66.10  PSAT = 6.50   QSTM = 0.85      TSAT = 143.89
STMLD =  257000.0           RSCALE = 0.00033

            **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 200. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  1.05 (FT.)
. . CONDENSER HEAT LOAD:    0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:    123.44 (DEG.F)
. . TOTAL NUMBER OF TUBES:  2096.0

. . EFFECTIVE TUBE LENGTH:             14.27 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:      5950. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:   41106. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:  997. (BTU/HR-FT**2-DEG.F)

LTOT = 16.13     LANE =  1.25      DTB =   3.60    ATB =  10.16
VTB = 145.03     DTS =  6.09       ATS =  29.10

         **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =   429.7   WMISC = 11093.3   LHW =  5.17     DPTHDR =  3.04
```

| | MATERIAL DENSITY(LB/FT**3) | THICKNESS (IN.) | WEIGHT (LB.) |
|---|---|---|---|
| OUTER TUBE SHEET----- | 558.0 | 6.0000 | 13421. |
| INNER TUBE SHEET----- | 488.0 | 2.0000 | 3912. |
| TUBE SUPPORT PLATE--- | 488.0 | 0.5625 | 1708. |
| TUBE BUNDLE---------- | 282.0 | *.**** | 5616. |
| CONDENSER SHELL------ | 488.0 | 0.6250 | 7907. |
| HOTWELL-------------- | 488.0 | 0.5625 | 3599. |
| WATERBOX------------- | 558.0 | 2.6400 | 7781. |

```
. . TOTAL CONDENSER DRY WEIGHT=   55466.6 (LB.)
. . TOTAL CONDENSER WET WEIGHT=   63449.9 (LB.)
                                  28.33 (TON)

. . TOTAL CONDENSER HEIGHT=  24.34 (FT.)
. . OUTER SHELL DIAMETER=     6.19 (FT.)
```

```
. . ENCLOSED BOX VOLUME=    1002.2 (FT**3)
    DRYCG =  12.33    WETCG =  11.73

. . CONDENSER FRICTIONAL HEAD LOSS=  84.1 (FT.)
. . TOTAL SYSTEM HEAD LOSS=         137.1 (FT.)
. . TOTAL SYSTEM PUMPING POWER=     381.3 (HP)
```

```
                    ******************************
                       CONDENSER NUMBER(  2)
                    ******************************
. . CONDENSER TYPE: SUBMARINE DESIGN
. . DOUBLE PASS CONDENSER WITH "U-TUBE" CONSTRUCTION.
. . DOUBLY ENHANCED TUBES . .
                          **** DATA INPUT ****
. . FLOW RATE:  7900. (GPM)  . . FLOW VELOCITY: 10.00 (FT/SEC) . .
DI = 0.555    DW = 0.625    AE = 0.010    AEPE = 0.2450    NTYPE =  2
EI = 0.0240   PI = 0.0949   FE =  4.54    MI = 0.580       RI = 0.0075
KW =   9.60   TCI =  66.10  PSAT = 6.50   QSTM = 0.85      TSAT = 143.89
STMLD =  257000.0           RSCALE = 0.00033

          **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 200. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  1.05 (FT.)
. . CONDENSER HEAT LOAD:    0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:    123.44 (DEG.F)
. . TOTAL NUMBER OF TUBES:  2096.0

. . EFFECTIVE TUBE LENGTH:             14.27 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:      5950. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:    41106. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:   997. (BTU/HR-FT**2-DEG.F)

LTOT = 16.13     LANE =  1.25     DTB =   3.60    ATB =  10.16
VTB = 145.03     DTS =  6.09      ATS =  29.10

       **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =   429.7   WMISC = 10614.2   LHW =  5.17    DPTHDR =  3.04
```

| | MATERIAL DENSITY(LB/FT**3) | THICKNESS (IN.) | WEIGHT (LB.) |
|---|---|---|---|
| OUTER TUBE SHEET----- | 558.0 | 6.0000 | 13421. |
| INNER TUBE SHEET----- | 488.0 | 2.0000 | 3912. |
| TUBE SUPPORT PLATE--- | 488.0 | 0.5625 | 1708. |
| TUBE BUNDLE---------- | 282.0 | *.**** | 6868. |
| CONDENSER SHELL------ | 488.0 | 0.6250 | 7907. |
| HOTWELL-------------- | 488.0 | 0.5625 | 3599. |
| WATERBOX------------- | 558.0 | 2.6400 | 4612. |

```
. . TOTAL CONDENSER DRY WEIGHT=   53071.1 (LB.)
. . TOTAL CONDENSER WET WEIGHT=   61054.4 (LB.)
                                  27.26 (TON)

. . TOTAL CONDENSER HEIGHT=  24.34 (FT.)
. . OUTER SHELL DIAMETER=     6.19 (FT.)
```

```
. . ENCLOSED BOX VOLUME=    1002.2 (FT**3)
    DRYCG =  12.00    WETCG =  11.41

. . CONDENSER FRICTIONAL HEAD LOSS=  89.9 (FT.)
. . TOTAL SYSTEM HEAD LOSS=         142.9 (FT.)
. . TOTAL SYSTEM PUMPING POWER=     397.3 (HP)
```

```
                    ******************************
                        CONDENSER NUMBER(  3)
                    ******************************
. . CONDENSER TYPE: SUBMARINE DESIGN
. . SINGLE PASS CONDENSER . .
. . DOUBLY ENHANCED TUBES . .
                         **** DATA INPUT ****
. . FLOW RATE: 13000. (GPM)  . . FLOW VELOCITY: 10.00 (FT/SEC) . .
DI = 0.555    DW = 0.625    AE = 0.010    AEPE = 0.2450    NTYPE =  2
EI = 0.0240   PI = 0.0949   FE =  4.54    MI = 0.580       RI = 0.0075
KW =   9.60   TCI =  66.10  PSAT = 6.50   QSTM = 0.85      TSAT = 143.89
STMLD =  257000.0           RSCALE = 0.00033

          **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 200. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  1.05 (FT.)
. . CONDENSER HEAT LOAD:    0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:    100.95 (DEG.F)
. . TOTAL NUMBER OF TUBES:  1724.0

. . EFFECTIVE TUBE LENGTH:            13.09 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:     4981. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:  33785. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:  928. (BTU/HR-FT**2-DEG.F)

LTOT = 14.94     LANE =  1.38      DTB =   3.26    ATB =   8.36
VTB = 109.37     DTS =  6.02       ATS =  28.42


        **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =   389.7   WMISC = 10193.1   LHW =  5.17     DPTHDR =  3.01
```

| | MATERIAL DENSITY(LB/FT**3) | THICKNESS (IN.) | WEIGHT (LB.) |
|---|---|---|---|
| OUTER TUBE SHEET----- | 558.0 | 6.0000 | 13540. |
| INNER TUBE SHEET----- | 488.0 | 2.0000 | 3947. |
| TUBE SUPPORT PLATE--- | 488.0 | 0.5625 | 1421. |
| TUBE BUNDLE---------- | 282.0 | *.**** | 4279. |
| CONDENSER SHELL------ | 488.0 | 0.6250 | 7239. |
| HOTWELL-------------- | 488.0 | 0.5625 | 3556. |
| WATERBOX------------- | 558.0 | 2.6400 | 6400. |

```
. . TOTAL CONDENSER DRY WEIGHT=   50965.3 (LB.)
. . TOTAL CONDENSER WET WEIGHT=   58689.8 (LB.)
                                  26.20 (TON)

. . TOTAL CONDENSER HEIGHT=  23.12 (FT.)
. . OUTER SHELL DIAMETER=     6.12 (FT.)
```

```
. . ENCLOSED BOX VOLUME=      934.4 (FT**3)
    DRYCG =  10.97    WETCG =  10.46

. . CONDENSER FRICTIONAL HEAD LOSS=  43.1 (FT.)
. . TOTAL SYSTEM HEAD LOSS=          78.9 (FT.)
. . TOTAL SYSTEM PUMPING POWER=     361.0 (HP)
```

```
                   ******************************
                      CONDENSER NUMBER(  4)
                   ******************************
. . CONDENSER TYPE: SUBMARINE DESIGN
. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HEADER DESIGN.
. . DOUBLY ENHANCED TUBES . .
                         **** DATA INPUT ****
. . FLOW RATE:  7900. (GPM)  . . FLOW VELOCITY: 10.00 (FT/SEC) . .
DI = 0.555    DW = 0.625    AE = 0.010    AEPE = 0.2450    NTYPE =  2
EI = 0.0240   PI = 0.0949   FE =  4.54    MI = 0.580       RI = 0.0075
KW =   9.60   TCI =  66.10  PSAT = 6.50   QSTM = 0.85      TSAT = 143.89
STMLD =  257000.0           RSCALE = 0.00033

             **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 300. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  2.25 (FT.)

* * PROGRAM WARNING NR-3: SPECIFIED CONDENSATE DRAINAGE PLATE SPACING IS
BEYOND TUBE SUPPORT LIMIT * * MAXIMUM TUBE DEFLECTION=  0.480 (IN.)
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION LIMITS - SECTION (
. . CONDENSER HEAT LOAD:     0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:     123.44 (DEG.F)
. . TOTAL NUMBER OF TUBES:  2096.0

. . EFFECTIVE TUBE LENGTH:             14.45 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:      5688. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:   36119. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:  954. (BTU/HR-FT**2-DEG.F)

LTOT = 16.30     LANE =  0.82      DTB =   3.60    ATB =  10.16
VTB = 146.81     DTS =  5.23       ATS =  21.52


          **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =   429.7   WMISC =  9128.7   LHW =  5.20     DPTHDR =  2.62
```

| | MATERIAL DENSITY(LB/FT**3) | THICKNESS (IN.) | WEIGHT (LB.) |
|---|---|---|---|
| OUTER TUBE SHEET----- | 558.0 | 6.0000 | 9187. |
| INNER TUBE SHEET----- | 488.0 | 2.0000 | 2678. |
| TUBE SUPPORT PLATE--- | 488.0 | 0.5625 | 788. |

| | | | |
|---|---|---|---|
| TUBE BUNDLE---------- | 282.0 | *.**** | 5677. |
| CONDENSER SHELL------ | 488.0 | 0.6250 | 6882. |
| HOTWELL-------------- | 488.0 | 0.5625 | 3092. |
| WATERBOX------------- | 558.0 | 2.6400 | 7781. |

. . TOTAL CONDENSER DRY WEIGHT= 45643.3 (LB.)
. . TOTAL CONDENSER WET WEIGHT= 50770.8 (LB.)
22.67 (TON)

. . TOTAL CONDENSER HEIGHT= 24.13 (FT.)
. . OUTER SHELL DIAMETER= 5.34 (FT.)
. . ENCLOSED BOX VOLUME= 748.3 (FT**3)
DRYCG = 12.37 WETCG = 11.56

. . CONDENSER FRICTIONAL HEAD LOSS= 85.0 (FT.)
. . TOTAL SYSTEM HEAD LOSS= 138.0 (FT.)
. . TOTAL SYSTEM PUMPING POWER= 383.7 (HP)

```
                 ****************************
                     CONDENSER NUMBER(  5)
                 ****************************
. . CONDENSER TYPE: SURFACE SHIP DESIGN
. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HEADER DESIGN.
. . DOUBLY ENHANCED TUBES . .
                         **** DATA INPUT ****
. . FLOW RATE:  7900. (GPM)  . . FLOW VELOCITY: 10.00 (FT/SEC) . .
DI = 0.555    DW = 0.625    AE = 0.010    AEPE = 0.2450    NTYPE =  2
EI = 0.0240   PI = 0.0949   FE =  4.54    MI = 0.580       RI = 0.0075
KW =   9.60   TCI =  66.10  PSAT = 6.50   QSTM = 0.85      TSAT = 143.89
STMLD =  257000.0           RSCALE = 0.00033

           **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 200. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  1.05 (FT.)
. . CONDENSER HEAT LOAD:    0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:    123.44 (DEG.F)
. . TOTAL NUMBER OF TUBES:  2096.0

. . EFFECTIVE TUBE LENGTH:             14.27 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:      5950. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:    41106. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:  997. (BTU/HR-FT**2-DEG.F)

LTOT = 14.61      LANE =  1.25      DTB =   3.60    ATB =  10.16
VTB = 145.03      DTS =  6.09       ATS =  29.10


          **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =   429.7   WMISC =  6579.4   LHW =  5.17     DPTHDR =  3.04
```

| | MATERIAL DENSITY(LB/FT**3) | THICKNESS (IN.) | WEIGHT (LB.) |
|---|---|---|---|
| TUBE SHEET----------- | 488.0 | 2.0000 | 3912. |
| TUBE SUPPORT PLATE--- | 488.0 | 0.5625 | 1708. |
| TUBE BUNDLE---------- | 282.0 | *.**** | 5086. |
| CONDENSER SHELL------ | 488.0 | 0.6250 | 7161. |
| HOTWELL-------------- | 488.0 | 0.5625 | 3599. |
| WATERBOX------------- | 558.0 | 1.5000 | 4421. |

```
. . TOTAL CONDENSER DRY WEIGHT=   32896.9 (LB.)
. . TOTAL CONDENSER WET WEIGHT=   40877.5 (LB.)
                                  18.25 (TON)

. . TOTAL CONDENSER HEIGHT=  22.82 (FT.)
. . OUTER SHELL DIAMETER=     6.19 (FT.)
. . ENCLOSED BOX VOLUME=     943.9 (FT**3)
```

```
    DRYCG =  11.18    WETCG =  10.40

. . CONDENSER FRICTIONAL HEAD LOSS=  76.7 (FT.)
. . TOTAL SYSTEM HEAD LOSS=         129.7 (FT.)
. . TOTAL SYSTEM PUMPING POWER=     360.8 (HP)
```

```
                    ******************************
                       CONDENSER NUMBER(  6)
                    ******************************
. . CONDENSER TYPE: SURFACE SHIP DESIGN
. . SINGLE PASS CONDENSER . .
. . SMOOTH INTERNAL TUBES . .
                          **** DATA INPUT ****
. . FLOW RATE: 13000. (GPM)  . . FLOW VELOCITY:  8.00 (FT/SEC) . .
DI = 0.555    DW = 0.625    AE = 0.010    AEPE = 0.2450    NTYPE =  1
EI = 0.0000   PI = 0.0000   FE =  4.54    MI = 0.000       RI = 0.0000
KW =   9.60   TCI =  66.10  PSAT = 6.50   QSTM = 0.85      TSAT = 143.89
STMLD =  257000.0           RSCALE = 0.00033

          **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 200. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  1.65 (FT.)
. . CONDENSER HEAT LOAD:     0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:     100.95 (DEG.F)
. . TOTAL NUMBER OF TUBES:  2155.0

. . EFFECTIVE TUBE LENGTH:           15.94 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:    1419. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:  37557. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:  594. (BTU/HR-FT**2-DEG.F)

LTOT = 16.27     LANE =  1.10      DTB =   3.65    ATB =  10.45
VTB = 166.52     DTS =  5.84       ATS =  26.82


       **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =   435.7   WMISC =  6692.9   LHW =  5.15     DPTHDR =  2.92
```

| | MATERIAL DENSITY(LB/FT**3) | THICKNESS (IN.) | WEIGHT (LB.) |
|---|---|---|---|
| TUBE SHEET----------- | 488.0 | 2.0000 | 3517. |
| TUBE SUPPORT PLATE--- | 488.0 | 0.5625 | 1349. |
| TUBE BUNDLE---------- | 282.0 | *.**** | 5826. |
| CONDENSER SHELL------ | 488.0 | 0.6250 | 7661. |
| HOTWELL-------------- | 488.0 | 0.5625 | 3438. |
| WATERBOX------------- | 558.0 | 1.5000 | 4545. |

```
. . TOTAL CONDENSER DRY WEIGHT=   33464.5 (LB.)
. . TOTAL CONDENSER WET WEIGHT=   40574.4 (LB.)
                                  18.11 (TON)

. . TOTAL CONDENSER HEIGHT=  24.35 (FT.)
. . OUTER SHELL DIAMETER=     5.95 (FT.)
. . ENCLOSED BOX VOLUME=     927.5 (FT**3)
```

```
    DRYCG =  11.51    WETCG =  10.69

. . CONDENSER FRICTIONAL HEAD LOSS=   6.6 (FT.)
. . TOTAL SYSTEM HEAD LOSS=          42.4 (FT.)
. . TOTAL SYSTEM PUMPING POWER=     194.0 (HP)
```

**** DATA SUMMARY ****

| CONDENSER NUMBER | SHELL DIAMETER (FT.) | TOTAL HEIGHT (FT.) | ENCLOSED BOX VOLUME (FT**3) | TOTAL WET WEIGHT (LB.) | TOTAL SYSTEM PUMPING POWER (HP) |
|---|---|---|---|---|---|
| 1. | 6.19 | 24.34 | 1002.2 | 63450. | 381.3 |
| 2. | 6.19 | 24.34 | 1002.2 | 61054. | 397.3 |
| 3. | 6.12 | 23.12 | 934.4 | 58690. | 361.0 |
| 4. | 5.34 | 24.13 | 748.3 | 50771. | 383.7 |
| 5. | 6.19 | 22.82 | 943.9 | 40878. | 360.8 |
| 6. | 5.95 | 24.35 | 927.5 | 40574. | 194.0 |

```
C*********************
C  PROGRAM: VERTCON1
C*********************
C PROGRAM "VERTCON1" IS A PRELIMINARY SIZING ROUTINE FOR VERTICAL TUBE
C STEAM SHIP/SUBMARINE PROPULSION CONDENSERS. THE PROGRAM TAKES INITIAL
C INPUT DATA BOTH INTERACTIVELY AND FROM DATA FILES. THE PROGRAM OUTPUT
C CONSISTS OF TABULATED VALUES FOR TOTAL & COMPONENT CONDENSER WEIGHTS,
C VOLUMES, AND PUMPING POWER.
C
C TWO BASIC VERTICAL TUBE CONFIGURATIONS ARE AVAILABLE WITH THIS PROGRAM
C AND THEY ARE: (1) DOUBLY ENHANCED TUBES, OR (2) SMOOTH INTERNAL TUBES
C WITH EXTERNAL ENHANCEMENT ONLY. SEVERAL GEOMETRIC VARIATIONS WITHIN
C THESE TWO BASIC CONFIGURATIONS CAN BE INVESTIGATED WITH "VERTCON1".
C
C ALSO, TWO BASIC CONSTRUCTION CONFIGURATIONS ARE AVAILABLE FOR "DOUBLE-
C PASS" CONDENSERS: (1) CONVENTIONAL RETURN HEADER DESIGN, AND
C                   (2) "U-TUBE" CONSTRUCTION
C
C THE VALUES OF CONDENSATE DRAINAGE PLATE SPACING (FT.), AND MAXIMUM
C MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC) MAY BE SPECIFIED BY THE
C USER INTERACTIVELY WHEN RUNNING THIS PROGRAM. IF THESE VALUES ARE
C NOT SPECIFIED, THEN THE PROGRAM WILL AUTOMATICALLY DETERMINE
C RECOMMENDED VALUES.
C
      DIMENSION DATA(100,5), OUTPUT(100,6)
      REAL ITSD,KSAT,KSW,KW,L1,LANE,LHW,LTOT,MI,MU,NT,NU
      INTEGER STM,CONFIG,CTYPE,DRAIN
      DATA NOE/'NO  '/,NO/'N   '/,ITYPE/'SUB '/
      DATA PIE/3.1415927/,PD/1.35/
C
C * * * STATEMENT FUNCTIONS FOR THERMAL & MECHANICAL PROPERTIES * * *
C
      NU(TCS)= .122181 - .1481615E-02 * TCS + .5516445E-05 * TCS**2
      RE(VM)=    VM * DI * 300. / NU(TCS)
      PRSW(TCS)= .0014369 * TCS**2 - .3134 * TCS + 21.89
      KSW(TCS)=  -.000003086 * TCS**2 + .001 * TCS + .291
C NU=   KINEMATIC VISCOSITY OF COOLING WATER (FT**2/HR)
C RE=   REYNOLDS NUMBER COOLING WATER
C PRSW= PRANDTL NUMBER COOLING WATER
C KSW=  CONDUCTIVITY OF COOLING WATER (BTU/HR-FT-DEG.F)
C
      FF(ARG2)= -1. / (2.46 * LOG(ARG2**MI + RI))
      ST(ARG3,ARG4)= FF(ARG2) / (5.68 * ARG3**(-.125) *SQRT(PRSW(TCS))
     1 * ARG4**.136 + GAMMA)
C FF= FRICTION FACTOR [SQRT(F/8)]
C ST= STANTON NUMBER
C
```

```
      KIN= 20
      KOUT= 21
      KSCR= 6
C KIN,KOUT, AND KSCR ARE OPERATORS THAT SPECIFY INPUT & OUTPUT DEVICES.
C IN THIS CASE KIN & KOUT SPECIFY INPUT & OUTPUT DATA FILES RESPECTIVELY
C AND KSCR SPECIFIES INPUT & OUTPUT AT COMPUTER TERMINAL.
C
C * * * * DATA(INTERACTIVE & FILE)-INPUT/OUTPUT * * * *
C
      WRITE(KSCR,100)
 100  FORMAT(1X,T14,'* * WELCOME TO PROGRAM: VERTCON,VERSION-1 * *'//T
     12,'ENTER PROGRAM RUN NUMBER')
      READ(KSCR,*) NUMBER
      WRITE(KOUT,104) NUMBER
 104  FORMAT(/////T21,'** PROGRAM: VERTCON,VERSION-1 **'////T25,'PROGR
     1AM RUN NUMBER(',I3,')'/////)
C
      WRITE(KOUT,106)
 106 FORMAT(1X,T26,'**** VARIABLE LIST ****'//T2,'AE= EXTERNAL FLUTE A
     1MPLITUDE (IN.)',T38,'.',T41,'MI= OPERAND FOR FRICTION FACTOR;'/T2,
     1'AEPE= AMPLITUDE-TO-PITCH RATIO OF',T38,'.',T45,'HELICAL INTERNAL
     1RIDGING'/T6,'EXTERNAL RIDGING',T38,'.',T41,'NTYPE= TUBE TYPE: 1-FO
     1R SMOOTH IN-'/T2,'ATB= AREA OF TUBE BUNDLE (FT**2)',T38,'.',T45,'T
     1ERNAL TUBE; AND 2-FOR DOUBLY'/T2,'ATS= AREA OF TUBE SHEET (FT**2)'
     1,T38,'.',T45,'ENHANCED TUBES'/T2,'DI= INTERNAL TUBE DIAMETER (IN.)
     1',T38,'.',T41,'PI= PITCH OF INTERNAL HELIX(IN.)'/T2,'DPTHDR= HEADE
     1R DEPTH (FT.)',T38,'.',T41,'PSAT= CONDENSER OPERATING PRESSURE'/T2
     1,'DRYCG= HEIGHT OF CENTER OF GRAVITY',T38,'.',T45,'(IN.HG)ABSOLUTE
     1'/T6,'ABOVE CONDENSER BOTTOM AT DRY',T38,'.',T41,'QSTM= QUALITY OF
     1 INLET STEAM TO'/T6,'WEIGHT (FT.)',T38,'.',T45,'CONDENSER'/T2,'DTB
     1= DIAMETER OF TUBE BUNDLE (FT.)',T38,'.',T41,'RI= OPERAND FOR FRIC
     1TION FACTOR,'/T2,'DTS= DIAMETER OF TUBE SHEET(FT.)',T38,'.',T45,'H
     1ELICAL INTERNAL RIDGING')
      WRITE(KOUT,108)
 108  FORMAT(1X,T2,'DW= TUBE WALL DIAMETER (IN.)',T38,'.',T41,'RSCALE=
     1 FOULING RESISTANCE'/T2,'EI= OPERAND FOR FRICTION FACTOR',T38,'.',
     1T45,'(HR-FT**2-DEG.F/BTU)'/T6,'HELICAL INTERNAL RIDGING',T38,'.',T
     141,'STMLD= STEAM LOAD (LBM/HR)'/T2,'FE= OPERAND FOR CONDENSING HEA
     1T',T38,'.',T41,'TSAT= SATURATION TEMP. (DEG.F)'/T6,'TRANSFER COEFF
     1ICIENT',T38,'.',T41,'VTB= VOLUME OF TUBE BUNDLE (FT**3)'/T2,'KW= C
     1ONDUCTIVITY OF TUBE WALL',T38,'.',T41,'WEXP= EXPANSION JOINT WEIGH
     1T (LB.)'/T6,'(BTU/HR-FT-DEG.F)',T38,'.',T41,'WMISC= WEIGHT OF MISC
     1ELLANEOUS'/T2,'LANE= STEAM LANE BREADTH (FT.)',T38,'.',T45,'COMPON
     1ENTS (LB.)'/T2,'LHW= LENGTH OF HOTWELL (FT.)',T38,'.',T41,'WETCG=
     1HEIGHT OF CENTER OF GRAVITY'/T2,'LTOT= TOTAL TUBE LENGTH (FT.)',T3
     18,'.',T45,'ABOVE CONDENSER BOTTOM AT WET'/T2,'TCI= INLET COOLANT T
     1EMP. (DEG.F)',T38,'.',T45,'WEIGHT (FT.)'/)
C
```

```
      WRITE(KSCR,110)
 110  FORMAT(1X,'ENTER THE NUMBER OF CONDENSERS TO BE SIZED')
      READ(KSCR,*) NCOND
C
      DO 10 N= 1, NCOND
C
      CONFIG= 3
      WRITE(KOUT,112)
 112  FORMAT('1',T21,'******************************')
      WRITE(KOUT,114) N
      WRITE(KSCR,114) N
 114  FORMAT(1X,T25,'CONDENSER NUMBER(',I3,')')
      WRITE(KOUT,115)
 115  FORMAT(1X,T21,'******************************')
C
      READ(KIN,121) CTYPE
 121  FORMAT(A4)
      READ(KIN,*) NTYPE,DI,DW,AE,AEPE,EI,PI
      READ(KIN,*) FLUTE,FE,MI,RI,KW,RSCALE
      READ(KIN,*) TCI,TSAT,QSTM,STMLD,PSAT,TIS
      READ(KIN,*) TOTS,TSHL,TTSP,THW,THDR,ITSD
      READ(KIN,*) OTSD,SHLD,TSPD,HWD,HDRD,TBD,ETUBE,TCOVER,COVERD
C
      IF(CTYPE .EQ. ITYPE) THEN
      INDEX1= 1
      WRITE(KOUT,138)
 138  FORMAT(1X,'. . CONDENSER TYPE: SUBMARINE DESIGN')
      ELSE
      INDEX1= 2
      WRITE(KOUT,140)
 140  FORMAT(1X,'. . CONDENSER TYPE: SURFACE SHIP DESIGN')
      END IF
C
      WRITE(KSCR,116)
 116  FORMAT(1X,'ENTER "1" FOR SINGLE PASS, OR "2" FOR DOUBLE PASS CON
     1DENSER')
      READ(KSCR,*) NPASS
      IF(NPASS .LE. 1) THEN
      WRITE(KOUT,118)
 118  FORMAT(1X,'. . SINGLE PASS CONDENSER . .')
      ELSE
      END IF
C
      IF(NPASS .GT. 1) THEN
      WRITE(KSCR,155)
 155  FORMAT(1X,'TWO BASIC CONFIGURATIONS ARE AVAILABLE FOR "DOUBLE PA
     1SS CONDENSERS: '/T5,'(1) CONVENTIONAL RETURN HEADER DESIGN, AND'/T
     15,'(2) "U-TUBE" CONSTRUCTION'/T2,'ENTER 1 OR 2 FOR CONFIGURATION S
```

```
     1ELECTION')
        READ(KSCR,*) CONFIG
        ELSE
        END IF
        IF(CONFIG .GT. 2) THEN
        GO TO 880
        ELSE IF(CONFIG .LE. 1) THEN
        WRITE(KOUT,160)
 160    FORMAT(1X,'. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HE
     1ADER DESIGN.')
        ELSE
        WRITE(KOUT,162)
 162  FORMAT(1X,'. . DOUBLE PASS CONDENSER WITH "U-TUBE" CONSTRUCTION.')
        END IF
 880    CONTINUE
C
        IF(NTYPE .LE. 1) THEN
        WRITE(KOUT,122)
 122    FORMAT(1X,'. . SMOOTH INTERNAL TUBES . .')
        ELSE
        WRITE(KOUT,124)
 124    FORMAT(1X,'. . DOUBLY ENHANCED TUBES . .')
        END IF
C
        WRITE(KSCR,126)
 126    FORMAT(1X,'ENTER: (1) COOLANT FLOW RATE (GPM)'/T9,'(2) COOLANT F
     1LOW VELOCITY (FT/SEC), AND'/T9,'(3) FRICTION FACTOR FOR INTERNAL T
     1UBE')
        READ(KSCR,*) GPM,VM,FFT
C
        WRITE(KOUT,128)
 128    FORMAT(1X,T27,'**** DATA INPUT ****')
        WRITE(KOUT,130) GPM,VM
 130    FORMAT(1X,'. . FLOW RATE: ',F6.0,' (GPM)  . . FLOW VELOCITY: ',F
     15.2,' (FT/SEC) . .')
        WRITE(KOUT,132) DI,DW,AE,AEPE,NTYPE,EI,PI,FE,MI,RI,KW,TCI,PSAT,
     1                  QSTM,TSAT,STMLD,RSCALE
 132    FORMAT(1X,'DI = ',F5.3,T16,'DW = ',F5.3,T30,'AE = ',F5.3,T44,'AE
     1PE = ',F6.4,T61,'NTYPE = ',I2/T2,'EI = ',F6.4,T16,'PI = ',F6.4,T30
     1,'FE = ',F5.2,T44,'MI = ',F5.3,T61,'RI = ',F6.4/T2,'KW = ',F6.2,T1
     16,'TCI = ',F6.2,T30,'PSAT = ',F4.2,T44,'QSTM = ',F4.2,T61,'TSAT =
     1',F6.2/T2,'STMLD = ',F9.1,T30,'RSCALE = ',F7.5/)
        WRITE(KOUT,136)
 136   FORMAT(1X,T15,'**** DATA OUTPUT - CONDENSER SIZING ROUTINE ****')
C
        WRITE(KSCR,142)
 142    FORMAT(1X,'DO YOU WANT TO SPECIFY MAXIMUM MAIN STEAM LANE ENTRAN
     1CE VELOCITY (FT/SEC)?'/T6,'YES OR NO')
```

```
      READ(KSCR,144) STM
 144  FORMAT(A4)
      IF((STM .EQ. NO) .OR. (STM .EQ. NOE)) THEN
      INDEX2= 2
      WRITE(KSCR,146)
 146  FORMAT(1X,'PROGRAM WILL SELECT RECOMMENDED MAXIMUM MAIN STEAM LA
     1NE ENTRANCE VELOCITY USING CONDENSER OPERATING PRESSURE AS SELECTI
     1ON CRITERIA.'//)
      ELSE
      INDEX2= 1
      WRITE(KSCR,148)
 148  FORMAT(1X,'ENTER MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/S
     1EC)')
      READ(KSCR,*) STMVEL
      END IF
C
      IF(INDEX2 .GT. 1) GO TO 770
      GO TO 771
 770  CONTINUE
      IF(PSAT .LT. 1.) THEN
      STMVEL= 500.
      ELSE IF(PSAT .GT. 5.) THEN
      STMVEL= 200.
      ELSE IF(PSAT .LE. 3.) THEN
      VEL= 600. - 100. * PSAT
      STMVEL= ANINT(VEL)
      ELSE IF(PSAT .GT. 3.) THEN
      VEL= 450. - 50. * PSAT
      STMVEL= ANINT(VEL)
      ELSE
      END IF
C STMVEL= MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC)
 771  CONTINUE
      WRITE(KOUT,150) STMVEL
 150  FORMAT(1X,'. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: ',F4.0
     1,' (FT/SEC)')
C
      WRITE(KSCR,164)
 164  FORMAT(1X,'DO YOU WANT TO SPECIFY CONDENSATE DRAINAGE PLATE SPAC
     1ING (FT.)?'/T6,'YES OF NO')
      READ(KSCR,166) DRAIN
 166  FORMAT(A4)
      IF((DRAIN .EQ. NO) .OR. (DRAIN .EQ. NOE)) THEN
      INDEX3= 2
      WRITE(KSCR,168)
 168  FORMAT(1X,'PROGRAM WILL SELECT RECOMMENDED CONDENSATE DRAINAGE P
     1LATE SPACING.'//)
      ELSE
```

```
      INDEX3= 1
      WRITE (KSCR,170)
 170  FORMAT(1X,'ENTER CONDENSATE DRAINAGE PLATE SPACING (FT.)')
      READ(KSCR,*) DRNPLT
      WRITE(KOUT,172) DRNPLT
 172  FORMAT(1X,'. . CONDENSATE DRAINAGE PLATE SPACING: ',F5.2,' (FT.)
     1'/)
      END IF
C
C * * * * FUNCTIONS FOR THERMAL & MECHANICAL PROPERTIES * * * *
C
      HFG=   1069.185 - 26.72568 * PSAT + 6.659357 * PSAT**2 -
     1       .9288262 * PSAT**3 + .05023536 * PSAT**4
      DENSC= PSAT**2 * .008829 - PSAT * .21843 + 62.347
      KSAT=  (PSAT**2 * (-.05904) + PSAT * 3.0435 + 356.6) * .001
      MU=    5.045709 - .0463856 * TSAT + .1242855E-03 * TSAT**2
      SIGMA= .0057445 - .7136956E-05 * TSAT - .64168E-08 * TSAT**2
      SPVOL= 1207.892 - 756.2292 * PSAT + 209.3 * PSAT**2 -
     1       25.8823 * PSAT**3 + 1.16875 * PSAT**4
C HFG=   LATENT HEAT OF CONDENSATION (BTU/LBM)
C DENSC= DENSITY OF CONDENSATE (LBM/FT**3)
C KSAT=  CONDUCTIVITY OF CONDENSATE (BTU/HR-FT-DEG.F)
C MU=    ABSOLUTE VISCOSITY OF CONDENSATE (LBM/HR-FT)
C SIGMA= SURFACE TENSION OF CONDENSATE (LBF/FT)
C SPVOL= SPECIFIC VOLUME OF STEAM AT SAT. CONDITIONS (FT**3/LBM)
C
      WF= 17875.5 / MU * DENSC**2 * AE**3 * AE/AEPE * EXP(-3.33 *AEPE)
C WF= FLOODING CONDENSATE FLOW RATE PER FLUTE (LB/HR)
C
      IF(AE .LE. .015) THEN
      SPEC1= 3.54272 - 522.1174 * AE + .1928462E+05 * AE**2
      ELSE IF (AE .LE. .025) THEN
      SPEC1= .3110063 - 25.2006 * AE + 520.0139 * AE **2
      ELSE
      SPEC1= .0303163 - 1.345982 * AE + 14.93315 * AE**2
      END IF
C
C * * * * CALCULATION OF BASIC CONDENSER DATA * * * *
C
      QT= STMLD * QSTM * HFG
      AW= GPM * .0022283 / VM
      WC= AW * VM * 230400.
C QT= CONDENSER HEAT LOAD (BTU/HR)
C AW= TOTAL CROSS-SECTIONAL AREA REQUIRED FOR COOLANT FLOW (FT**2)
C WC= COOLING WATER FLOW RATE (LB/HR)
C
      TOUT= TCI + QT / (.95 * WC)
      ARG1= DI / 24.
```

```
      NT=   AW / (PIE * ARG1**2)
      NT=   ANINT(NT)
      TNT=  NT * NPASS
      DN=   DW + AE * 2.
      OD=   DW + AE * 4.
C TOUT= OUTLET COOLANT TEMPERATURE (DEG.F)
C NT=   NUMBER OF TUBES PER CONDENSER PASS
C TNT=  TOTAL NUMBER OF TUBES
C DN=   NOMINAL TUBE DIAMETER (IN.)
C OD=   OUTSIDE TUBE DIAMETER (IN.)
C
C * * * * DETERMINATION OF CONDENSATE DRAINAGE PLATE SPACING * * * *
C
      MACH=  STMVEL / SQRT(3672.76 * (TSAT + 459.67))
      CDRAG= .75 * MACH + .975
      WDRAG= CDRAG * OD * DRNPLT * STMVEL**2 / (24. * SPVOL)
      PITCH= PD * OD
C MACH=  MACH NUMBER OF STEAM INLET FLOW TO CONDENSER
C CDRAG= DRAG COEFFICIENT FOR STEAM INLET FLOW OVER CONDENSER TUBES
C WDRAG= DRAG FORCE LOADING ON TUBES DUE TO STEAM VELOCITY (LBF.)
C PD=    TUBE PITCH-TO-DIAMETER RATIO
C PITCH= TUBE PITCH (IN.)
C
      IF(INDEX3 .LE. 1) GO TO 774
      DRNPLT= 2.5
C
      DO 40 N1= 1, 60
C
      DELT= 5.
      T2= 71.
C "DRNPLT=3." & "DELT=5." ARE BOTH ASSUMED VALUES TO INITIATE THE
C FOLLOWING ITERATIVE CALCULATIONS.
C
      DO 30 N2= 1, 190
C
      DTS= T2 - TCI
      TCS= TCI + DTS / 2.
      ARG7= (TSAT - TCI) / (TSAT - T2)
      DTLM= DTS / LOG(ARG7)
C DTLM= LOG MEAN TEMPERATURE DIFFERENCE (DEG.F)
C THIS ANALYSIS EXAMINES A SAMPLE TUBE SECTION OF LENGTH= DRNPLT TO
C DETERMINE RECOMMENDED VALUES FOR CONDENSATE DRAINAGE PLATE SPACING.
C THIS TUBE SECTION IS LOCATED IN THE COOLING WATER INLET SECTION OF THE
C CONDENSER, WHERE THE CONDENSATION RATE/PER UNIT TUBE LENGTH WILL BE AT
C A MAXIMUM.
C DRNPLT= CONDENSATE DRAINAGE PLATE SPACING (FT.)
C DELT=   SAT. TEMP. - TUBE WALL TEMP. (DEG.F)
C T2=     TUBE SECTION C.W. OUTLET TEMP. (DEG.F)
```

```
C DTS=    C.W. TEMP. RISE ACROSS TUBE SECTION (DEG.F)
C TCS=    AVGERAGE C.W. TEMP. FOR EACH SECTION (DEG.F)
C DTLM=   LOG MEAN TEMPERATURE DIFFERENCE (DEG.F)
C
        IF(NTYPE .LE. 1) THEN
        HW= .023 * KSW(TCS) / DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
C HW= HEAT TRANSFER COEFFICIENT COOLANT SIDE FOR SMOOTH INTERNAL TUBE
C     (BTU/HR-FT**2-DEG.F)
        ELSE
        ARG2= 7. / RE(VM)
        GAMMA= -1. * (LOG(2. * EI/DI) * 2.5 + 3.75)
        ARG3=  EI / PI
        ARG4=  EI / DI * RE(VM) * FF(ARG2)
        HW=    12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
C HW= HEAT TRANSFER COEFFICIENT COOLANT SIDE FOR DOUBLY ENHANCED TUBES
C     (BTU/HR-FT**2-DEG.F)
        END IF
C
        DO 20 N3= 1, 10
C
        ARG5= QSTM * HFG * WF / (DRNPLT * DELT)
        ARG6= KSAT**3 * DENSC*SIGMA*QSTM*HFG * 4.17*10.**8/(MU*DELT)
        ARG8= AE / 12.
        HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1         * ARG6**.2307
C HCOND= HEAT TRANSFER COEFFICIENT CONDENSATE (BTU/HR-FT**2-DEG.F)
C
        US= 1. / (DN/(DI*HW)+RSCALE + 1./HCOND + LOG(DN/DI)*DN/(24.*KW))
C US= OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR-FT**2-DEG.F)
C
        QS=   .95 * WC * DTS
        DELT= QS * 12. / (NT * PIE * DN * DRNPLT * HCOND)
C QS= HEAT LOAD FOR TUBE SECTION (BTU/HR)
  20    CONTINUE
C
        L1= QS * 12. / (NT * PIE * DN * US * DTLM)
        IF(L1 .GT. DRNPLT) THEN
        T2= T2 - .025
        ELSE
        T2= T2 + .049
        END IF
        COMP= ABS(L1 - DRNPLT)
        IF(COMP .LE. .01) THEN
        GO TO 772
        ELSE
        END IF
  30    CONTINUE
 772    CONTINUE
```

```
      WCWF= QS / (NT * QSTM* HFG * WF * FLUTE)
      DEFL= .26526*WDRAG*DRNPLT**3 / (ETUBE*(OD**4 - DI**4))
C DEFL= MAXIMUM TUBE DEFLECTION (IN.)
      IF(WCWF .LE. SPEC1) THEN
      WRITE(KOUT,176) DRNPLT
      WRITE(KSCR,176) DRNPLT
 176  FORMAT(1X,'. . CONDENSATE DRAINAGE PLATE SPACING: ',F5.2,' (FT.)')
      GO TO 773
      ELSE IF((WCWF .GT. SPEC1) .OR. (DEFL .GT. PITCH/2.)) THEN
      DRNPLT= DRNPLT - .05
      ELSE
      END IF
  40  CONTINUE
      IF(COMP .GT. .05) THEN
      WRITE(KSCR,174)
      WRITE(KOUT,174)
 174  FORMAT(1X,'* * PROGRAM WARNING NR-1: "DRNPLT" BEYOND SPECIFICATI
     1ON LIMITS * *'/)
      ELSE
      END IF
 773  CONTINUE
      IF(WCWF .GT. SPEC1) THEN
      WRITE(KSCR,178)
      WRITE(KOUT,178)
 178  FORMAT(1X,'* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION
     1 LIMITS * *'/)
      ELSE
      END IF
 774  CONTINUE
      DEFL= 1375. * WDRAG * DRNPLT**3 / (ETUBE * (OD**4 - DI**4))
      IF(DEFL .GT. PITCH/2.) THEN
      WRITE(KOUT,180) DEFL
      WRITE(KSCR,180) DEFL
 180  FORMAT(1X,'* * PROGRAM WARNING NR-3: SPECIFIED CONDENSATE DRAINA
     1GE PLATE SPACING IS BEYOND TUBE SUPPORT LIMIT * *'/T2,'. . MAXIMUM
     1 TUBE DEFLECTION= ',F6.3,' (IN.)')
      ELSE
      END IF
C * * * * CALCULATION OF EFFECTIVE TUBE LENGTH * * * *
C
      SUML=  0.0
      SUMQ=  0.0
      AVGHW= 0.0
      AVGHC= 0.0
      AVGU=  0.0
      T1= TCI
C
      DO 70 N4= 1, 100
```

```
C
      DELT= 5.
      T2=   T1 + 5.
C
      DO 60 N5= 1, 190
C
      DTS=  T2 - T1
      TCS=  T1 + DTS / 2.
      ARG7= (TSAT - T1) / (TSAT - T2)
      DTLM= DTS / LOG(ARG7)
C
      IF(NTYPE .LE. 1) THEN
      HW= .023 * KSW(TCS)/DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
      ELSE
      ARG2= 7. / RE(VM)
      GAMMA= -1. * (LOG(2. * EI/DI) * 2.5 + 3.75)
      ARG3= EI / PI
      ARG4= EI / DI * RE(VM) * FF(ARG2)
      HW= 12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
      END IF
C
      DO 50 N6= 1, 10
C
      ARG5= QSTM * HFG * WF / (DRNPLT * DELT)
      ARG6= KSAT**3 * DENSC * SIGMA * QSTM * HFG*4.17*10.**8/(MU*DELT)
      ARG8= AE / 12.
      HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1        * ARG6**.2307
C
      US= 1. / (DN/(DI*HW)+RSCALE + 1./HCOND +LOG(DN/DI)*DN/(24.*KW))
      QS= .95 * WC * DTS
      DELT= QS * 12. / (NT * PIE * DN * DRNPLT * HCOND)
  50  CONTINUE
C
      L1= QS * 12. / (NT * PIE * DN * US * DTLM)
      IF(L1 .GT. DRNPLT) THEN
      T2= T2 - .025
      ELSE
      T2= T2 + .049
      END IF
C
      COMP= ABS(L1-DRNPLT)
      IF(COMP .LE. .01) THEN
      GO TO 775
      ELSE
      END IF
  60  CONTINUE
C
```

```
 775    WCWF= QS / (NT * QSTM * HFG * WF * FLUTE)
        IF(WCWF .GT. SPEC1) THEN
        WRITE(KOUT,182) N4
        WRITE(KSCR,182) N4
 182    FORMAT(1X,'* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION
     1 LIMITS - SECTION (',I3,') * * ')
        ELSE
        END IF
C
        DATA(N4,1)= QS
        DATA(N4,2)= HW
        DATA(N4,3)= HCOND
        DATA(N4,4)= US
        DATA(N4,5)= DRNPLT
C * * CALCULATION OF FINAL SECTION LENGTH * *
        SUML=  DATA(N4,5) + SUML
        SUMQ=  DATA(N4,1) + SUMQ
        AVGHW= DATA(N4,2) + AVGHW
        AVGHC= DATA(N4,3) + AVGHC
        AVGU=  DATA(N4,4) + AVGU
        DELT= 5.
        QFS= QT - SUMQ
        TCS= (TOUT + T2)/2.
C
        IF(NTYPE .LE. 1) THEN
        HW= .023 * KSW(TCS)/DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
        ELSE
        ARG2= 7./ RE(VM)
        ARG4= EI / DI * RE(VM) * FF(ARG2)
        HW= 12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
        END IF
C
        TL= 2.0
        DO 80 N7= 1, 10
        ARG5=  QSTM * HFG * WF / (TL * DELT)
        ARG6= KSAT**3 * DENSC * SIGMA * QSTM * HFG*4.17*10.**8/(MU*DELT)
        HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1          * ARG6**.2307
        US= 1. /(DN/(DI*HW)+RSCALE + 1./HCOND + LOG(DN/DI)*DN/(24.*KW))
        ARG7= (TSAT - T2) / (TSAT - TOUT)
        DTLM= (TOUT - T2) / LOG(ARG7)
        QP=   QFS * 12. / (PIE * DN * NT)
        TL=   QP / (US * DTLM)
        DELT= QP / (TL * HCOND)
  80    CONTINUE
C
        IF(TL .LE. DRNPLT) THEN
        NS = N4 + 1
```

```
      SUML= SUML + TL
      AVGHW= AVGHW / NS
      AVGHC= AVGHC / NS
      AVGU=  AVGU  / NS
      GO TO 776
      ELSE
      T1= T2
      END IF
      WRITE(KSCR,181)N4
181   FORMAT(T2,'* CONDENSER SECTION(',I3,') CALCULATED *')
 70   CONTINUE
C
C * * CONDENSER WEIGHT & VOLUME CALCULATIONS * *
C
776   WRITE(KSCR,183)
183   FORMAT(T2,'* FINAL CONDENSER SECTION-CALCULATED *')
      TOTL= SUML / NPASS * 1.025
C
      IF(INDEX1 .LE. 1) THEN
      LTOT= (9.125 + TIS) / 6. + TOTL
      ELSE
      LTOT= TIS / 6. + TOTL
      END IF
C
      LANE= .000106 * STMLD * SPVOL / ((TOTL - 1.84) * STMVEL)
      TND=  166.27 / PITCH**2
C TOTL= TOTAL EFFECTIVE TUBE LENGTH (FT.)
C LTOT= TOTAL TUBE LENGTH (FT.)
C LANE= STEAM LANE BREADTH (FT.)
C TND=  NUMBER OF TUBES PER SQ. FT. FOR 60 DEGREE TRIANGULAR PITCH
C
      ATB=   TNT / TND
      DTB=   SQRT(4. * ATB / PIE)
      ARG10= (DTB + 2. * LANE) / 2.
      ATS=   PIE * ARG10**2
      DTS=   DTB + 2 * LANE
      VTB=   TOTL * ATB
C ATB= AREA OF TUBE BUNDLE (FT**2)
C DTB= DIAMETER OF TUBE BUNDLE (FT.)
C ATS= AREA OF TUBE SHEET (FT**2)
C DTS= DIAMETER OF TUBE SHEET (FT.)
C VTB= VOLUME OF TUBE BUNDLE (FT**3)
C
      ARG12= DTS - 2 * LANE + OD / 6.
      WP1=   .785398 * (ARG12**2 - OD**2 * TNT * .00694)
      WP2=   .785398 * (DTS**2 - OD**2 *TNT* .00694)
      WOTS=  OTSD * TOTS / 6. * WP2
      WITS=  ITSD * TIS / 6. * WP2
```

```
      WTSP=  NS / NPASS * TSPD * TTSP/12. * WP1
      WTB=   .005454 * (DN**2 - DI**2) * LTOT * TBD * TNT
      DSHL=  DTS + TSHL / 6.
      DPTHDR= DTS / 2.
      WSHL=  .785398 * SHLD * LTOT * (DSHL**2 -DTS**2)
C WOTS= WEIGHT OF OUTER TUBE SHEETS (LB.)
C WITS= WEIGHT OF INNER TUBE SHEETS (LB.)
C WTSP= WEIGHT OF TUBE SUPPORT PLATES (LB.)
C WTB=  WEIGHT OF TUBE BUNDLE (LB.)
C WSHL= WEIGHT OF SHELL (LB.)
C DSHL= OUTER DIAMETER OF SHELL (FT.)
C DPTHDR= HEADER DEPTH (FT.)
C
      WEXP= .39167 * DTB * TSHL * SHLD
      WHDR= .4083 * DTB**2 * THDR * HDRD
      VHW=  STMLD / 3690. - .0417 * ATB * NTSP - .2618 * TOTL
      DHW=  DSHL + 1.
      LHW=  VHW / (.7854 * DHW**2) + DPTHDR + .5
      WHW=  .2618 * (DHW**2 / 4. + DHW * LHW) * THW * HWD
C LHW=  LENGTH OF HOTWELL (FT.)
C VHW=  VOLUME OF HOTWELL (FT.)
C WEXP= WEIGHT OF EXPANSION JOINT (LB.)
C WHDR= WEIGHT OF HEADER (LB.)
C WHW=  WEIGHT OF HOTWELL (LB.)
C
      IF(CONFIG .GT. 2) THEN
      GO TO 999
      ELSE
      END IF
      IF(CONFIG .GT. 1) THEN
      WCOVER= .20415 * DTB**2 * TCOVER * COVERD
      WHDR= WHDR/2. + WCOVER
      WUTUBE= .005454 * (DN**2 - DI**2) * DTB * TBD * TNT
      WTB=  .005454 * (DN**2 - DI**2) * LTOT * TBD * TNT + WUTUBE
      ELSE
      END IF
 999  CONTINUE
      WT=    WOTS + WITS + WTSP + WTB + WSHL + WEXP + WHDR + WHW
      WMISC= WT * .25
      WDRY=  WT * 1.25
      WLIQ=  .005454 * DI**2 *NT*LTOT + 33.51 * DSHL**3 +STMLD/226935.
      WET=   WDRY + WLIQ
      TONS=  WET / 2240.
C WMISC= WEIGHT OF MISCELLANEOUS COMPONENTS (LB.)
C WDRY=  TOTAL DRY WEIGHT OF CONDENSER (LB.)
C WLIQ=  TOTAL LIQUID WEIGHT (LB.)
C WET=   TOTAL WET WEIGHT OF CONDENSER (TONS)
C
```

```
      CONDH=   LHW + DPTHDR + LTOT
      BOXVOL=  DSHL**2 * (LTOT + DPTHDR) + DHW**2 * LHW
C CONDH=  CONDENSER HEIGHT (FT.)
C BOXVOL= CONDENSER ENCLOSED BOX VOLUME (FT**3)
C
      ARG14= (LHW + LTOT/2.) * (WOTS + WITS + WTSP + WSHL + WEXP)
     1        + (LHW/2.) * WHW + CONDH/2. * WMISC
      IF(CONFIG .GT. 1) THEN
      DRYCG= (ARG14 +(LHW - DPTHDR/2.)*WDHDR/2. + (LHW+LTOT+DPTHDR/2.)
     1  * (WCOVER + WUTUBE) + (LHW + LTOT/2.) * WTB) / WDRY
      ELSE
      DRYCG= (ARG14 + (LHW + LTOT/2.) * (WTB + WHDR)) / WDRY
      END IF
      ARG15= STMLD / 3690. * 61.5
      ARG16= WLIQ - ARG15
      WETCG= (DRYCG * WDRY + ARG16 * (LHW + LTOT/2.) + ARG15 * VHW /
     1        25.1328) / WET
C DRYCG=HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM FOR CONDENSER
C       AT DRY WEIGHT (FT.)
C WETCG=HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM FOR CONDENSER
C       AT WET WEIGHT (FT.)
C
      IF(CONFIG .GT. 1) THEN
      HT=  FFT * (LTOT + DTB/2.) * NPASS * VM**2 / (DI * 5.353)
      HWB= (1.5 + .42 * (VM - 8.)) / 2.
      HE=  (1.2 + .6  * (VM - 6.)) / 2.
      ELSE
      HT=  FFT * LTOT * NPASS * VM**2 / (DI * 5.353)
      HWB= 1.5 + .42 * (VM - 8.)
      HE=  1.2 + .6  * (VM - 6.)
      END IF
C
      HLP= 79.7 - .00338 * GPM
C HT=  FRICTIONAL HEAD LOSS FOR TUBES (FT.)
C HWB= WATERBOX INLET & OUTLET LOSSES (FT.)
C HE=  TUBE END LOSSES (FT.)
C HLP= SEAWATER CIRCULATING LOOP PIPING LOSSES (FT.)
C
      TOTHL= HT + HWB + HE
      SYSL=  TOTHL + HLP
      PMPWR= .000352 * GPM * SYSL
C TOTHL= TOTAL FRICTIONAL HEAD LOSS FOR CONDENSER (FT.)
C SYSL=  TOTAL FRICTIONAL HEAD LOSS FOR CONDENSER AND SYSTEM PIPING(FT.)
C PMPWR= TOTAL SYSTEM PUMPING POWER (HP)
C
      A= N
      OUTPUT(N,1)= A
      OUTPUT(N,2)= DSHL
```

```
      OUTPUT(N,3)= CONDH
      OUTPUT(N,4)= BOXVOL
      OUTPUT(N,5)= WET
      OUTPUT(N,6)= PMPWR
C
C * * * * * CALCULATION RESULTS - OUTPUT(DATA FILE) * * * *
      WRITE(KOUT,190) QT,TOUT,TNT
 190  FORMAT(1X,'. . CONDENSER HEAT LOAD:   ',E12.5,' (BTU/HR)'/T2,'.
     1. OUTLET COOLANT TEMP:     ',F6.2,' (DEG.F)'/T2,'. . TOTAL NUMBER O
     1F TUBES: ',F7.1/)
      WRITE(KOUT,192) TOTL,AVGHW,AVGHC,AVGU
 192  FORMAT(1X,'. . EFFECTIVE TUBE LENGTH:',T39,F5.2,' (FT.)'/T2,'. .
     1 AVG. HEAT TRNFR. COEFF. C.W.:',T38,F6.0,' (BTU/HR-FT**2-DEG.F)'/T
     12,'. . AVG. HEAT TRNFR. COEFF. COND.:',T38,F6.0,' (BTU/HR-FT**2-DE
     1G.F)'/T2,'. . AVG. OVERALL HEAT TRNFR. COEFF.:',T38,F6.0,' (BTU/HR
     1-FT**2-DEG.F)'/)
      WRITE(KOUT,194) LTOT,LANE,DTB,ATB,VTB,DTS,ATS,WEXP,WMISC,LHW,
     1                DPTHDR
 194  FORMAT(1X,'LTOT = ',F5.2,T19,'LANE = ',F5.2,T37,'DTB = ',F6.2,T5
     13,'ATB = ',F6.2/T2,'VTB = ',F6.2,T19,'DTS = ',F5.2,T37,'ATS = ',F6
     1.2///T12,'**** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****'//T
     12,'WEXP = ',F7.1,T19,'WMISC = ',F7.1,T37,'LHW = ',F5.2,T53,'DPTHDR
     1 = ',F5.2///T30,'MATERIAL',T47,'THICKNESS',T64,'WEIGHT'/T26,'DENSI
     1TY(LB/FT**3)',T49,'(IN.)',T65,'(LB.)')
      IF(INDEX1 .GT. 1) THEN
      WRITE(KOUT,195) ITSD,TIS,WITS
 195  FORMAT(T2,'TUBE SHEET-----------',T31,F5.1,T49,F6.4,T64,F6.0)
      ELSE
      WRITE(KOUT,196) OTSD,TOTS,WOTS,ITSD,TIS,WITS
 196  FORMAT(T2,'OUTER TUBE SHEET-----',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1           'INNER TUBE SHEET-----',T31,F5.1,T49,F6.4,T64,F6.0)
      END IF
      WRITE(KOUT,197) TSPD,TTSP,WTSP,TBD,WTB,SHLD,TSHL,WSHL,HWD,THW,
     1                WHW,HDRD,THDR,WHDR,WDRY
 197  FORMAT(T2,'TUBE SUPPORT PLATE---',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1       'TUBE BUNDLE----------',T31,F5.1,T49,'*.****',T64,F6.0/T2,
     1           'CONDENSER SHELL------',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1           'HOTWELL--------------',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1           'WATERBOX-------------',T31,F5.1,T49,F6.4,T64,F6.0//T2
     1          ,'. . TOTAL CONDENSER DRY WEIGHT=  ',F8.1,' (LB.)')
      WRITE(KOUT,198) WET,TONS,CONDH,DSHL,BOXVOL,DRYCG,WETCG,TOTHL,
     1 SYSL,PMPWR
 198  FORMAT(1X,'. . TOTAL CONDENSER WET WEIGHT=  ',F8.1,' (LB.)'/T35,
     1F6.2,' (TON)'//T2,'. . TOTAL CONDENSER HEIGHT= ',F6.2,' (FT.)'/T2,
     1'. . OUTER SHELL DIAMETER=   ',F6.2,' (FT.)'/T2,'. . ENCLOSED BOX
     1VOLUME=   ',F7.1,' (FT**3)'/T6,'DRYCG = ',F6.2,T24,'WETCG = ',F6.2
     1//T2,'. . CONDENSER FRICTIONAL HEAD LOSS= ',F5.1,' (FT.)'/T2,'. .
     1TOTAL SYSTEM HEAD LOSS=',T38,F5.1,' (FT.)'/T2,'. . TOTAL SYSTEM PU
```

```
     1MPING POWER=',T38,F5.1,' (HP)'///)
  10    CONTINUE
        WRITE(KOUT,200)
 200    FORMAT('1',T26,'**** DATA SUMMARY ****'//T4,'CONDENSER',T16,'SHE
     1LL',T27,'TOTAL',T36,'ENCLOSED',T48,'TOTAL WET',T61,'TOTAL SYSTEM'/
     1T5,'NUMBER',T15,'DIAMETER',T26,'HEIGHT',T35,'BOX VOLUME',T49,'WEIG
     1HT',T60,'PUMPING POWER'/T16,'(FT.)',T27,'(FT.)',T36,'(FT**3)',T50,
     1'(LB.)',T64,'(HP)')
C
        DO 90 N8= 1, NCOND
        WRITE(KOUT,202)(OUTPUT(N8,J),J=1,6)
 202    FORMAT(T6,F3.0,T15,F6.2,T26,F6.2,T36,F7.1,T49,F7.0,T63,F5.1)
  90    CONTINUE
        STOP
        END
```

# APPENDIX B

## CONTENTS

# TABLE 5

## Properties of Multiple-Helix Internal Ridged Tubes

| Symbol | Tube no. | Nominal diameter, in | Root diameter $d_r$, in | Fin count, fins/in | Outside area, $ft^2/ft$ | ID (max) $d_i$, in | No. of starts |
|---|---|---|---|---|---|---|---|
| △ | 12 | $\frac{3}{4}$ | 0.623 | (Stripped) | 0.163 | 0.573 | 5 |
| ▷ | 30 | $\frac{3}{4}$ | 0.627 | 26.4 | 0.640 | 0.575 | 6 |
| ▽ | 22 | $\frac{3}{4}$ | 0.625 | 26 | 0.640 | 0.569 | 6 |
| ◁ | 27 | $\frac{3}{4}$ | 0.622 | 26.5 | 0.640 | 0.572 | 6 |
| ▲ | 28 | $\frac{3}{4}$ | 0.624 | 26 | 0.640 | 0.573 | 5 |
| ▶ | 31 | $\frac{3}{4}$ | 0.624 | 26.6 | 0.640 | 0.576 | 6 |
| ▼ | 29 | $\frac{3}{4}$ | 0.625 | 26.1 | 0.640 | 0.575 | 5 |
| ◀ | 38 | $\frac{3}{4}$ | 0.633 | 38.0 | 0.830 | 0.572 | 8 |
| ◇ | 37 | $\frac{3}{4}$ | 0.624 | 38.5 | 0.901 | 0.574 | 10 |
| | 40 | $\frac{3}{4}$ | 0.627 | 27.3 | 0.689 | 0.561 | 10 |
| ○ | 41 | $\frac{3}{4}$ | 0.628 | 38.1 | 0.852 | 0.572 | 12 |
| ◁ | 9 | $\frac{3}{4}$ | 0.628 | 38.5 | 0.901 | 0.575 | 10 |
| □ | 21 | $\frac{7}{8}$ | 0.740 | 26 | 0.640 | 0.684 | 6 |
| ■ | 19 | $\frac{7}{8}$ | 0.745 | (Stripped) | 0.195 | 0.692 | 6 |
| | 44 | $\frac{3}{4}$ | 0.627 | 41.0 | | 0.574 | 6 |
| | 43 | $\frac{3}{4}$ | 0.628 | 41.0 | | 0.573 | 10 |
| ○ | 42 | $\frac{3}{4}$ | 0.626 | 41.0 | 0.901 | 0.573 | 10 |
| | 46 | $\frac{3}{4}$ | 0.627 | 41.2 | | 0.573 | 10 |
| | 45 | $\frac{3}{4}$ | 0.627 | 41.2 | | 0.577 | 10 |
| ○ | 13 | 1 | 0.883 | 26 | 0.841 | 0.820 | 6 |
| ⊖ | 32 | 1 | 0.877 | 26 | 0.871 | 0.825 | 6 |
| ◓ | 25 | 1 | 0.880 | 26 | 0.841 | 0.816 | 6 |
| ⊙ | 24 | 1 | 0.878 | 26 | 0.841 | 0.814 | 6 |
| ◒ | 23 | 1 | 0.880 | - 26 | 0.841 | 0.816 | 6 |
| ◐ | 26 | 1 | 0.864 | 27.3 | 0.841 | 0.815 | 6 |

| Symbol | Internal ridging | | Internal aspect ratios | | | ID of envelope tube, in |
|---|---|---|---|---|---|---|
| | Height $e$, in | Pitch $p$, in | $e/d_i$ | $e/p$ | $p/d_i$ | |
| △ | 0.0125 | 0.475 | 0.0218 | 0.0263 | 0.829 | 1.000 |
| ▷ | 0.0162 | 0.279 | 0.0282 | 0.0581 | 0.485 | 1.000 |
| ▽ | 0.0165 | 0.320 | 0.0288 | 0.0516 | 0.558 | 1.000 |
| ◁ | 0.017 | 0.385 | 0.0297 | 0.0442 | 0.673 | 1.000 |
| ▲ | 0.019 | 0.475 | 0.0332 | 0.0400 | 0.829 | 1.000 |
| ▶ | 0.0198 | 0.287 | 0.0344 | 0.0690 | 0.498 | 1.000 |
| ▼ | 0.0207 | 0.469 | 0.0360 | 0.0441 | 0.816 | 1.000 |
| ◀ | 0.018 | 0.212 | 0.0315 | 0.0849 | 0.371 | 1.000 |
| ◇ | 0.0200 | 0.166 | 0.0348 | 0.1205 | 0.289 | 1.000 |
| | 0.0175 | 0.170 | 0.0312 | 0.1029 | 0.303 | 1.000 |
| ○ | 0.0215 | 0.138 | 0.0376 | 0.1558 | 0.241 | 1.000 |
| ◁ | 0.0204 | 0.191 | 0.0355 | 0.1070 | 0.332 | 1.000 |
| □ | 0.017 | 0.391 | 0.0249 | 0.0435 | 0.572 | 1.2348 |
| ■ | 0.017 | 0.285 | 0.0246 | 0.0596 | 0.412 | 1.000 |
| | 0.021 | 0.207 | 0.0366 | 0.101 | 0.361 | – |
| | 0.021 | 0.124 | 0.0366 | 0.169 | 0.216 | – |
| ○ | 0.024 | 0.0949 | 0.0419 | 0.253 | 0.166 | 1.000 |
| | 0.021 | 0.094 | 0.0367 | 0.223 | 0.164 | – |
| | 0.015 | 0.094 | 0.0260 | 0.1596 | 0.163 | – |
| ○ | 0.0178 | 0.333 | 0.0217 | 0.0535 | 0.406 | 1.5936 |
| ⊖ | 0.0193 | 0.335 | 0.0234 | 0.0576 | 0.406 | 1.5936 |
| ◓ | 0.0205 | 0.340 | 0.0253 | 0.0603 | 0.420 | 1.5936 |
| ⊙ | 0.0205 | 0.330 | 0.0252 | 0.0621 | 0.405 | 1.5936 |
| ◒ | 0.0205 | 0.338 | 0.0251 | 0.0607 | 0.414 | 1.5936 |
| ◐ | 0.021 | 0.340 | 0.0258 | 0.0618 | 0.417 | 1.5936 |

TABLE 6 Friction Factor Characteristics of Multiple-Helix Internal Ridged Tubes

| Tube no. | $m$ | $r$ |
|---|---|---|
| 12 | 0.762 | 0 |
| 30 | 0.64 | −0.00039 |
| 22 | 0.697 | −0.00017 |
| 27 | 0.72 | −0.00026 |
| 28 | 0.72 | −0.00028 |
| 31 | 0.61 | −0.00095 |
| 29 | 0.68 | −0.00064 |
| 38 | 0.61 | −0.00109 |
| 37 | 0.54 | −0.00392 |
| 40 | 0.627 | −0.00080 |
| 41 | 0.57 | −0.00259 |
| 9 | 0.59 | −0.00197 |
| 21 | 0.70 | −0.00014 |
| 19 | 0.626 | 0 |
| 44 | 0.53 | −0.00180 |
| 43 | 0.55 | +0.00017 |
| 42 | 0.58 | +0.00750 |
| 46 | 0.54 | +0.00275 |
| 45 | 0.52 | −0.00159 |
| 13 | 0.63 | +0.00024 |
| 32 | 0.70 | +0.00090 |
| 25 | 0.645 | +0.00032 |
| 24 | 0.64 | −0.00035 |
| 23 | 0.637 | +0.00028 |
| 26 | 0.64 | +0.00018 |

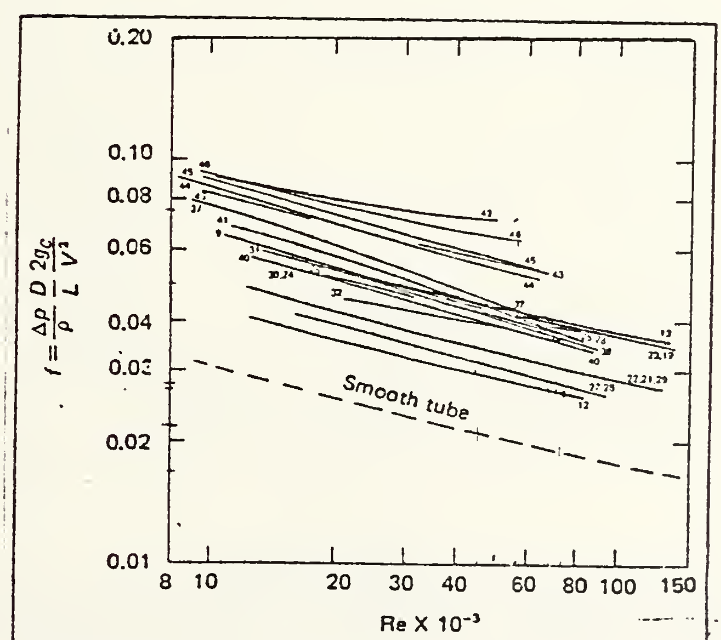



FIGURE 11 Friction Factor Curves for Multiple-Helix Internal Ridged Tubes

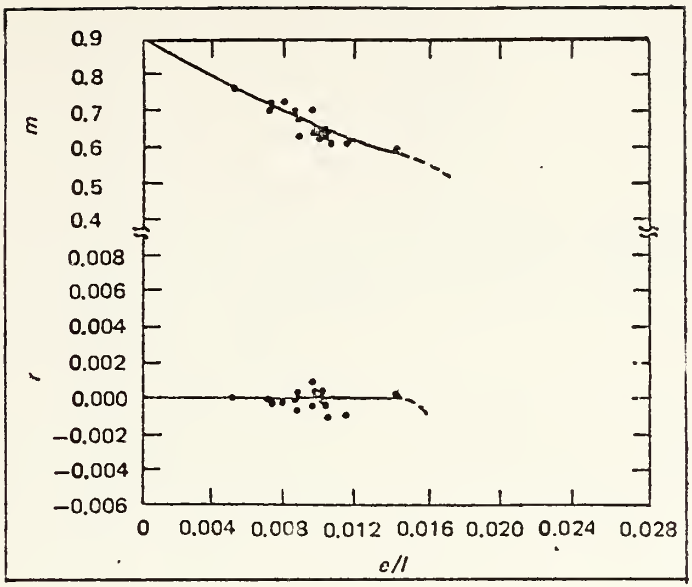



FIGURE 12a
Operands for friction factor equation versus geometric aspect ratio e/l for multiple-helix internal ridging with $p/d_i \geq 0.36$

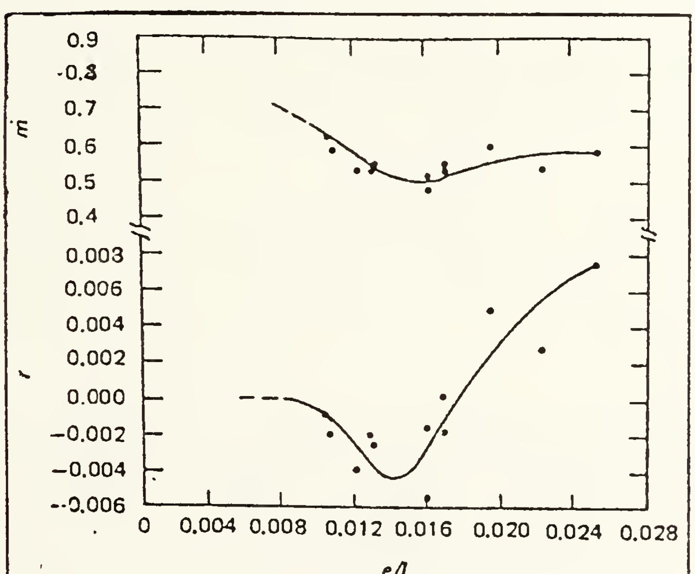



FIGURE 12b
Operand for friction factor equation versus geometric aspect ratio e/l for multiple-helix internal ridging with $p/d_i < 0.36$

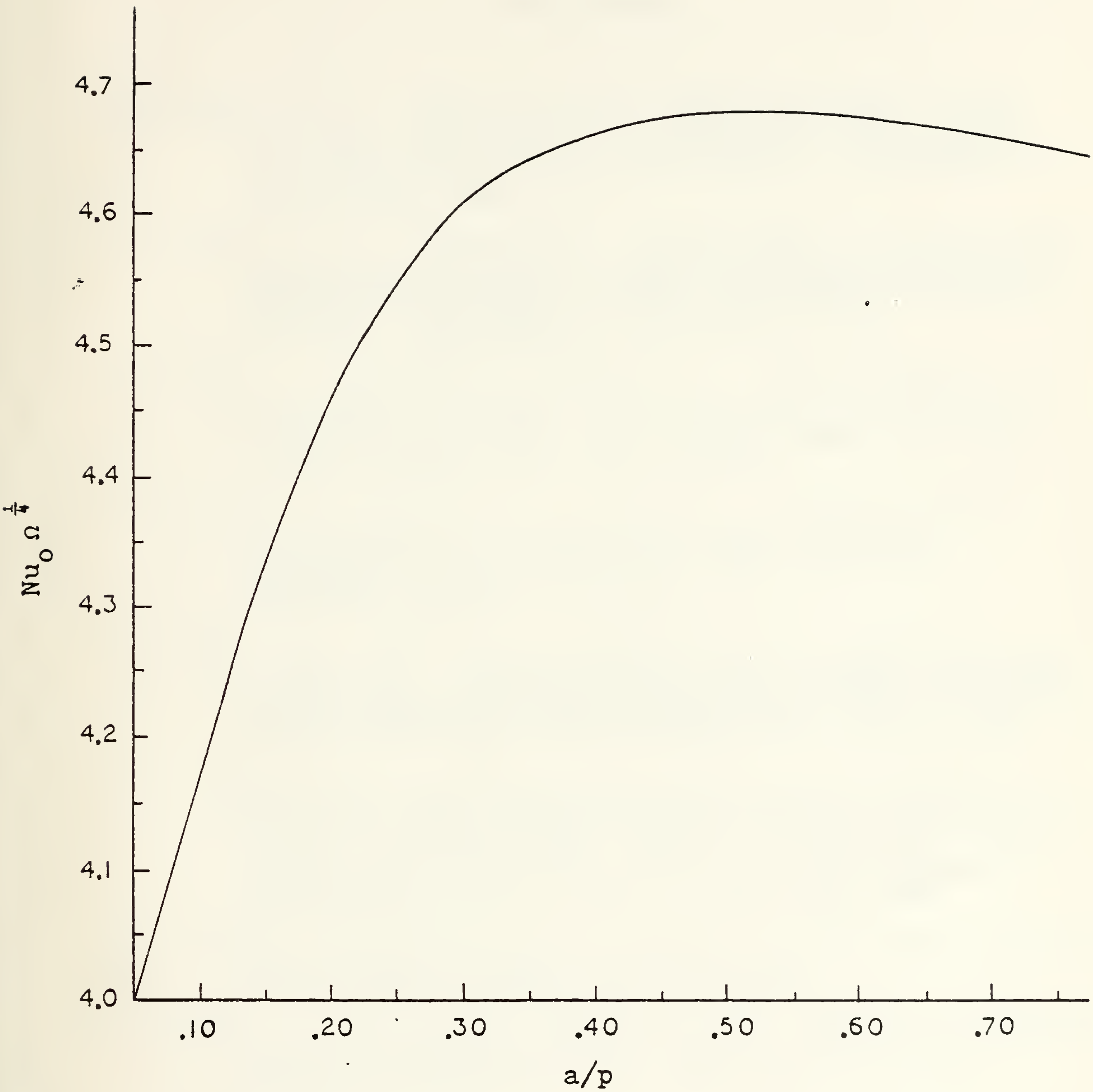



FIGURE 13

$Nu_c\Omega^{\frac{1}{4}}$ vs. a/p for Axial Fluted Vertical Tubes

## REFERENCES


(1) Barnes, C.G., "Marine Surface Condenser Design Using Vertical Tubes Which Are Enhanced," O.E. Thesis in Ocean Engineering, Massachusetts Institute of Technology, 1981.

(2) Michel, J.W., "Discussion of Enhancement in Condensers," Proceedings of the Workship on Modern Developments in Marine Condensers, Naval Postgraduate School, Monterey, California, March 26-28, 1980.

(3) Mori, Y., Hyikata, K., Hirasawa, S., and Nakayama, W., "Optimized Performance of Condensers with Outside Condensing Surface," Condensation Heat Transfer, ASME, New York, 1979, pp. 55-62.

(4) Panchal, C. and Bell, K., "Analysis of Nusselt-Type Condensation on a Vertical Fluted Surface," Condensation Heat Transfer, ASME, New York, 1979, pp. 45-54.

(5) Webb, R.L., "The Use of Enhanced Surface Geometries in Condensers," Proceedings of the Workshop on Modern Developments in Marine Condensers, Naval Postgraduate School, Monterey, California, March 26-28, 1980.

(6) Withers, J.G., "Tube-Side Heat Transfer and Pressure Drop for Tubes Having Helical Internal Ridging with Turbulent/Transitional Flow of Single-Phase Fluid, Part 2. Multiple-Helix Ridging," Heat Transfer Engineering, Vol. 2, No. 2, Oct.-Dec. 1980.

(7) Military Specification, "Condensers, Steam, Naval Shipboard," MIL-C-15430J (Ships) 19 June 1974.

(8) Neuber, H., Theory of Notch Stresses, Principles for Exact Stress Calculation, J. W. Edwards, Ann Arbor, Michigan, 1946.

(9) Harrington, R., ed., Marine Engineering, Society of Naval Architects and Marine Engineers, New York, 1971.

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c.1 Performance comparison of enhanced steam condensers.